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Field Evaluation of a Solar Energy Intensifier-Thermal Energy Storage System

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FIELD EVALUATION
OF A
SOLAR ENERGY INTENSIFIER-THERMAL ENERGY STORAGE
SYSTEM

BY

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SYSTEM

This thesis is approved as a creditable and independent investigation by a candidate for the degree, Master of Science, and is acceptable for meeting the thesis requirements for this degree. Acceptance of this thesis does not imply that the conclusions reached by the candidate are necessarily the conclusions of the major department.

Thesis Advisor — Date

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TABLE OF CONTENTS

Chapter

Page

LIST OF TABLES

Table

Page

LIST OF FIGURES

Figure

Page

INTRODUCTION

Solar energy has been recognized as a potential source of energy for low-temperature agricultural applications. Many processes, such as space heating, drying of farm products or waste materials, and water heating require low-temperature heat which can be supplied by solar radiation. Hellickson, et. al. (1981a) listed eight circumstances which make solar energy particularly attractive to the agricultural industry. Four of these circumstances were:

1. Adequate land area,
2. Numerous processes requiring low temperature rise,
3. Compatibility between low temperature rise and high solar system efficiency, resulting in lower cost, and
4. Availability of the necessary air moving equipment as part of existing crop drying and livestock ventilation systems.

Research was begun at South Dakota State University in 1976 on a practical solar system for agricultural applications. Investigation centered around a low-cost, multi-use system which was portable and required a minimum of maintenance. The present design evolved through extensive testing and evaluation of several design configurations and various construction materials. The system was designed to be in use year-round for crop drying, ventilation air preheating, and water heating so that the economic return on the investment could be maximized.

Performance of the solar system was tested in laboratory and field conditions for short durations. However, long term performance of the SEI-TES system in a commercial application had not been documented. Clearly, the most reliable performance ratings belong to those systems which have undergone extended testing in the field. Therefore research was initiated with the following objectives:

1. Evaluate and document the performance of the SEI-TES system under actual ventilation air preheating conditions at an operating livestock production facility.
2. Compare conventional fuel requirements in a solar-assisted facility with those in a similar non-solar facility.
3. Identify problems that result from actual field use, and recommend design changes which will improve the effectiveness of the system.
4. Recommend design parameters for sizing systems to be installed in ventilation heating applications.
5. Compare solar-system costs with the value of conventional fuel saved.

REVIEW OF LITERATURE

Solar Engineering Principles

According to Kreith and Kreider (1978), approximately 1.7×10^{14} kilowatts of solar radiation is incident upon the earth. Approximately 30% of the intercepted energy is reflected to space, low-temperature heat which is reradiated to space makes up 47% of the total, 23% of the radiant energy fuels the evaporation/precipitation cycle, and less than 0.5% of the incident energy is evident as kinetic energy in wind and waves or stored in plants through photosynthesis.

Kreith and Kreider (1978) also stated that only about one third of the total radiation intercepted by the earth actually reaches the surface. Seventy percent of the surface radiation falls on the oceans, while around 1.5×10^{17} kilowatt-hours reaches land in a year. Only a small portion of the radiant energy can be utilized because of physical, social, and economic limitations.

Available solar energy for a given location can be determined from insolation records and solar climate maps from a number of sources. Many of the best sources of climatic information were listed by Lytle and Hellickson (1980). However, Brewer, et. al. (1981) cautioned that data for many areas are not precise, and that unusual atmospheric conditions can contribute to substantial differences between calculated and actual radiation values.

In many areas where insolation is plentiful and relatively uninterrupted, there is very little demand for supplemental heat for space heating or crop drying. Conversely, in regions having colder, damp climates, the demand for supplemental heat is high, but the insolation levels are not high enough to be economically utilized (Brewer, et. al., 1981).

Collectors

According to Patton (1975), the primary objective of the collector is to convert solar radiation into useful heat. Kreith and Kreider (1978) stated that this conversion is accomplished by a surface facing the sun which absorbs a portion of the incident solar energy and transfers it to a working fluid in contact with the surface. The most common form of solar collector is the flat-plate collector. Generally, the flat-plate collector is a simple, durable device which can collect low-to-moderate-temperature solar energy efficiently. The flat-plate collector is economically useful for space heating, air conditioning, and domestic hot-water heating (Patton, 1975).

Duffie and Beckman (1980) stated that the two critical components of the flat-plate collector are the cover glazing and the absorber, since the transmittance of the cover and the absorptivity and emissivity of the absorber determine how much of the incident radiation will actually be available to the transfer fluid.

The most common absorber materials are copper, aluminum (Trotter, Heid, and McElroy, 1979), and sheet steel (Duffie and Beckman, 1980). Inexpensive flat black coatings can be used to enhance the thermal absorption of the absorber materials. These coatings may have an absorptivity up to 0.98 (Midwest Plan Service, 1978). However, the emissivity of the surface is related to the absorptivity, and, according to Patton (1975), reducing the losses from infrared emissivity with more expensive selective coatings results in lowered absorptivity values. Patton (1975) also suggests disadvantages for both flat-black paints and selective surfaces. The less expensive black paint may become faded or cracked during the long-term life of a collector while dust and moisture in harsh environments may have a very adverse effect on delicate selective surfaces. Brewer, et. al. (1981) recommended flat-black paint for low-temperature agricultural applications since the selective coatings made a significant improvement only in relatively high-temperature situations.

Patton's survey (1975) of cover materials revealed that glass was the most suitable glazing for flat-plate collectors because of its high transmissivity in the solar spectrum and its opaqueness to long-wave radiation emitted outward from the absorber. Patton (1975) also found that many of the polyvinyl and plastic glazings tested were much less durable than glass and had undesirable thermal expansion properties. Brewer, et. al. (1981) recommended single glazing for low-temperature flat-plate collectors using air as the working fluid because of the minimal convective heat losses at low temperatures.

The selection of a working fluid with optimal characteristics is important. Kreith and Kreider (1978) listed several advantages of air-type collectors. These included: no freezing problems, little internal corrosion, leaks have less serious consequences, and no heat exchanger is needed between the collector, storage, and building. These same advantages make air a suitable transfer medium for agricultural applications (Brewer, et. al., 1981).

Because the convective heat-transfer coefficient for air is considerably lower than it is for water, Kreith and Kreider (1978) recommend providing the largest possible surface-to-air contact area for maximum performance. This can be effectively accomplished for low-temperature, relatively high air flow applications by bringing the incoming air between the glazing and the absorber and forcing it to return behind the absorber (Patton, 1975). This double-pass pattern reduces losses by keeping the warmest air away from the glazing while doubling the effective heat transfer area of the absorber. It is also essential to maintain turbulent flow conditions in air collectors to obtain maximum thermal performance (Holman, 1981).

Orientation is another factor in flat plate collector design. Brewer, et. al. (1981) noted that, although the best orientation for a collector in the northern hemisphere would be south-facing with an angle of tilt equal to the latitude minus the solar declination, the constant adjustment necessary for seasonal variations of declination would not be feasible for most installations. They recommended an angle of tilt equal to the latitude for year-round use and latitude plus or minus 15 degrees for primarily winter or summer use, respectively.

Concentrating Reflectors

The terms reflector and concentrator are used interchangeably throughout the literature. Kreith and Kreider (1978) defined concentrators as reflecting or refracting devices which are oriented so that the incident radiation is focused onto a receiver, thus increasing the surface flux intensity above that normally striking the receiver area alone. Duffie and Beckman (1980) and Meinel and Meinel (1976) present detailed optical analyses of the most common concentrating collectors.

The most prevalent reason for incorporating concentrators in solar system designs is to increase the operating temperature for high temperature applications (Duffie and Beckman, 1980). Hellickson (1980) stated that concentrators can also be used to reduce solar system costs by reducing the collector surface area while maintaining the desired system output.

Reflector orientation is of considerable importance. Duffie and Beckman (1980) noted that high-temperature, point-focusing concentrators must have sophisticated tracking systems to maintain a precise position relative to the sun. However, it was observed that for certain low-concentration, linear reflectors the optical requirements for focusing could be met by rotation about a single axis, and that, for some designs, adjustments about this axis were only necessary on a weekly or monthly basis. Kreith and Kreider (1978) mentioned the existence of a group of collectors which employ flat-plate absorbers combined with reflectors which increase radiation flux by factors of 2 to 10. This intermediate group did not produce a sharp focus and therefore required neither accurate tracking mechanisms nor expensive high-precision optical components.

A wide range of reflective materials are available for use with concentrating collectors (Solar Age Catalog, 1977). Collins, et. al. (1983) suggested that metalized polyester films may be the most cost-effective reflective material for low-temperature, linear concentrators.

An inherent disadvantage of concentrators, expressed by Kreith and Kreider (1978), is that only a small portion of the diffuse energy incident upon them can be effectively deflected to the receiver surface.

Thermal Storage

Patton (1975) stated that for space heating applications, a thermal storage medium must be provided in order to retain energy collected during periods of high insolation for use on cloudy days or at night. According to Kreith and Kreider (1978), the additional cost and complexity of thermal storage is necessary to provide heat storage capacity to reduce fluctuations in available energy.

The following are listed by Patton (1975) and Kreith and Kreider (1978) as important factors to consider in selecting storage material: physical relationship to the building, cost of the thermal storage, which includes both the container and the storage material, storage capacity per unit volume, and cycling life of the material. Kreith and Kreider (1978) divided storage materials into two classes: latent heat storage and sensible heat storage. For low temperature applications such as space heating, they recommended sensible heat storage in the form of water or rocks. According to Patton (1975), rock and gravel were used in a number of prototype solar systems because of simplicity, reliability, and low maintenance. Another advantage of using rocks with air-heating collectors is that the need for a heat exchanger is eliminated (Kreith and Kreider, 1978).

Brewer, et. al. (1981) listed three methods for the designer to use in limiting heat losses from storage. These were: minimize the surface area-to-volume ratio, use the appropriate amount of insulation, and place storage below ground or in the building to be heated to reduce the temperature difference between the storage and its surroundings.

Storage bed shape is another important facet of the design. Kreith and Kreider (1978) discussed the need to consider tradeoffs between pressure drop and fluid residence time in order to design a system that is both economical and effective.

Design Considerations

Kreith and Kreider (1978) stated that, "It is apparent that the economic viability of solar energy improves with increases in collector efficiency, insolation level, alternative fuel cost, and total life of the equipment, but diminishes with increasing initial cost of the conversion equipment." In light of this conclusion, Duffie and Beckman (1980) suggested that there may be merit in designing collectors with a lower efficiency than is actually possible if a significant reduction in cost is achieved.

For agricultural applications, designers must put top priority on durability, simplicity, and ease of care and maintenance. The basic design and choice of materials should minimize breakage of glazing, deterioration of absorber coating, moisture damage due to condensation or leaking, dust and debris on collector and storage surfaces, and corrosion of the structural material and ductwork (Brewer, et. al., 1981).

Climate is the single most important factor in deciding whether or not a particular solar system is feasible in a given location. Climate influences the amount of radiation available and is a major cause of deterioration. Therefore, Brewer, et. al. (1981) noted that designs must be well planned on the basis of a careful study of local weather patterns.

The probability of hail must be considered in determining the type of glazing material to use, according to Brewer, et. al. (1981). They also note that while rain may be sufficient to remove dust from some residential and commercial collectors, it may not remove contaminants commonly found around grain drying and livestock housing facilities. Snow covering the collector surface was listed as another potential source of inefficiency.

Patton (1975) urged careful design of glazing for ease in replacing broken panels. Patton also suggested protection against dust, moisture, and other foreign material which can enter an air-heating collector and seriously affect heat collection and transfer capabilities.

The behavior of collector materials at abnormally high temperatures is an important aspect of solar-system design. Patton (1975) remarked that such a situation might occur as a consequence of a power outage, resulting in a loss of fluid circulation. Duffie and Beckman (1980) also stressed the fact that the collector materials must be able to withstand high temperatures, and the design must accomodate the thermal expansion which is likely to occur under no-flow conditions.

Both Duffie and Beckman (1980), and Patton (1975), recommended designing for an average effective system life of about twenty years. Due to the high initial cost, a solar system must be able to operate for an extended period with relatively low maintenance costs in order to realize a payback.

Evaluation of Performance

According to Kreith and Kreider (1978), a disproportionately small number of performance evaluations were available to designers for air-cooled collectors. Part of the reason for this lack of information was the assumption that reasonably accurate predictions of air-cooled flat-plate collector performance could be made with basic heat transfer principles (Kreith and Kreider, 1978). Duffie and Beckman (1980) stated that while there is no basic reason for working collectors to perform differently than predicted, there are certain conditions which can cause deviations from expected performance. Four sources of discrepancies were listed:

1. Nonuniform fluid flow through portions of the collector,
2. Fluid flow rates which depart significantly from the design flow rate; because of the heavy dependence of the heat-transfer rate on flow rate, flow deviations can seriously affect performance,
3. Leaks in air-type collectors, and
4. Edge and back losses which are dependent on the size and geometry of the collector unit.

Kreith and Kreider (1978) cited five additional reasons for diminished performance under actual operating conditions:

1. Optical efficiency decreases as surface incidence angles increase,
2. High ratios of diffuse-to-beam-radiation decrease optical efficiency,
3. Dust and foreign matter on the outer glazing reduce transmissivity
4. Final energy output may be reduced by transient effects, and
5. Deterioration of surface coatings reduces potential for energy absorption.

The most common measure of solar system performance is the collection efficiency, defined by Duffie and Beckman (1980) as the time-integrated collector output divided by the total available solar energy integrated over the same time period. The relationship is represented by

$$N = \int Q\, dt / A \int I\, dt$$

where

N= efficiency

Q= useful energy gain

dt= differential time period

A= collector area

I= solar energy available

Kreith and Kreider (1978) presented the following energy balance as a basis for evaluating collector performance:

$$(I)(A)(t)(a) = q_u + q_l + de/dt$$

where

I= incident solar radiation,

A= area of collector,

t= mean transmissivity of the cover,

a= surface absorptivity,

q_u = useful energy delivered,

q_l = energy loss due to conduction, convection, and radiation, and

de/dt= rate of internal energy storage.

Lytle and Hellickson (1980) presented a comprehensive study of the climatic and system parameters necessary for proper evaluation of agricultural solar systems. In that study, a standardized set of parameters was developed for easy comparison of the operating characteristics of a given group of solar system designs. Lytle and Hellickson listed temperatures, temperature rises, efficiencies, energy collected, energy utilized, and economic values as the critical information to be included in a report of system performance. Other important parameters included flow rates, dimensions, materials, and climatic data.

Economics are of great importance to designers since, in a free economy, the basis for widespread acceptance of a concept is not only technological reality, but economic competitiveness as well (Kreith and Kreider, 1978). Hellickson (1979) developed a method of combining performance results with climatic information and economic factors to predict the economic feasibility of agricultural solar collectors. Application of this method makes it possible for designers to construct performance curves for a given combination of parameters which can then be compared to a "break-even" curve.

Standardized tests require the performance of collectors to be evaluated over a period of between fifteen and thirty minutes, after the system has reached a "quasi-static" condition (Hill and Kusuda, 1974). However, for design purposes, the performance over a full day or more is of more value (Kreith and Kreider, 1978).

While it is important to understand the operating characteristics of each system component, the characteristics of one element, the collector for example, cannot be used to predict the system energy output because of the many interactions involved. Kreith and Kreider (1978) declared that the most important measure of performance is the behavior of the entire integrated system. In the same way, peak ratings of components such as collectors and heat exchangers are inadequate for design purposes. The long-term performance of the entire system under all types of operating conditions is necessary for design decisions (Kreith and Kreider, 1978).

State of the Art Agricultural Applications

Kreith and Kreider (1978) advocated matching energy sources to tasks on the basis of entropy. Space heating, water heating, and crop drying at low temperatures were high-entropy tasks which were most effectively matched with low-temperature, high-entropy solar energy provided by flat-plate solar collectors. Brewer, et. al. (1981) concurred with the view that solar energy was well suited to low-temperature agricultural tasks. Applications requiring up to a maximum of 38 degrees Celsius temperature rise were listed as: general heating, grain and crop drying, heating livestock structures, heating greenhouses, processing, and irrigation.

Trotter, Heid, and McElroy (1979) found that the three most common agricultural uses being investigated were grain drying, livestock shelter heating, and water heating.

Grain Drying

Because of rapid improvement in harvesting techniques, Trotter, et. al., (1979) estimated that 65 to 70 percent of the corn crop, along with all rice, and 10 to 20 percent of the grain sorghum and soybeans, were mechanically dried in the U.S. This practice required the equivalent of 2.5 billion liters of liquefied petroleum gas per year. Heid and Trotter (1982) found that while solar energy was not cost-effective for high-speed, high-moisture applications, it could be used to supplement natural air for in-storage drying.

Shove (1981) reported the performance of simple collectors which were constructed as part of two separate machinery storage sheds. These collectors were incorporated as portions of the south wall (134 square meters) and roof (201 square meters) of each building. Heated air was drawn from the attic to nearby in-storage drying bins. The collectors provided a 24-hour average temperature rise of from 1 to 4 degrees Celsius during October and November, 1979. Twenty-four hour average power output ranged from 4 to 23 kilowatts. Total cost in 1976 for a collector of the type studied was $9.24 per square meter.

The use of a commercially produced Solair solar collector for in-storage drying of corn was investigated by Eno and Felderman (1980). A bin containing 24,900 kilograms of corn was dried from an initial moisture content of 22 percent to 13.5 percent over a 24-day period in October. The Solair heater, which consisted of evacuated absorber tubes fixed in the troughs of a series of wedge-shaped aluminum reflectors, added an average of 52,750 kilojoules of energy per day to the drying air. It was concluded that while the system performed well, the capabilities as well as the cost of the Solair heater far exceeded the requirements for natural air drying applications.

Chau and Baird (1980) studied the feasibility of solar grain drying under hot and humid conditions. A low-cost collector was constructed using a black plastic absorber covered with a clear plastic glazing and one or two plastic mesh screens suspended between the absorber and the glazing. Eighteen batches of corn and nine batches of soybeans were dried during a three-year test period. The collector achieved temperature rises ranging from 8 to 22 degrees Celsius with an air flow rate ranging from 5 to 16 cubic meters per minute per kilogram. Conclusions drawn from the study were that it was technically possible to dry grain with solar energy in hot, humid climates, however the depth of the grain must be less than one meter and the drying must be completed within one week. Because of the short drying season, it was found that solar drying was not competitive with fossil fuels, even though the collector cost was only $3.30 per square meter.

Rice drying with solar energy was the subject of a study by Calderwood (1980). Four different treatments were observed involving unheated air drying, solar heating directly with and without stirring, and solar heat applied indirectly from rock storage. Collectors were simple flat-plate designs using clear corrugated fiberglass glazing and corrugated steel roofing painted flat black for absorbers. The frames were made of 5 x 15 centimeter lumber. Calderwood concluded that drying time could be reduced with solar energy, thereby saving energy required to operate drying fans. There was no significant difference in the grain quality between the solar-heated and unheated air treatments. Average temperature rise was between 4 and 10 degrees Celsius for the direct solar heating over a ten-hour period, while the average temperature rise from the storage was between 3 and 11 degrees Celsius over a twenty-four-hour period. The recommended technique was direct heating of the drying air with no stirring of the grain.

A thorough economic analysis and simulation of solar-supplemented natural air drying of corn, based on the performance of a simple, portable, flat-plate collector was conducted by Heid and Aldis (1981). The analysis was based on temperature rises of 1.5, 3.2, and 4.8 degrees Celsius from the collector. The study concluded that supplemental solar drying in the west-central Great Plains was not economically feasible because system costs outweighed the losses due to deterioration of the corn.

Solar drying of large round hay bales has been studied. A flat plate roof collector tested by Bledsoe and Henry (1980) achieved a collection efficiency of 42 percent with an air flow rate of 0.5 cubic meters per minute per square meter of collector area. Bale drying in this experiment was enhanced by forcing air through holes punched in the centers of the bales. Morrison and Shove (1980) reported a 20 percent reduction in fan energy use for large round bales dried with solar heated air as opposed to unheated air drying.

Livestock Shelter Heating

According to Trotter, Heid, and McElroy (1979), solar heating for livestock has been concentrated in three areas: poultry brooding, swine farrowing, and dairy barn space heating. Because of the longer demand period, livestock shelter heating was seen as a potentially successful application of solar energy.

Brewer, et. al. (1981) asserted that a significant portion of the energy consumed in agricultural production is used for space heating poultry brooding facilities. The yearly energy requirement for brooding was given as 1.95×10^{13} kilojoules, most of which was supplied by liquefied petroleum gas.

Vertical wall collectors using single glazings of Kalwall plastic and black, building-felt absorbers provided ventilation air preheating in a brooding facility for a test by Flood, Koon, and Brewer (1980). The preheaters were used in conjunction with an array of double-glazed, flat-plate, water-heating collectors which were the main source of supplemental heating. During trials in all seasons, the solar system supplied 24 to 100 percent of the supplemental energy requirements in Alabama.

Drury, Mitchell, and Beard (1980) reported that three fourths of the required heating energy for brooding chickens in Georgia was supplied by an array of water-heating flat-plate collectors having a water thermal storage tank. Average efficiency of the collectors was 33 percent, with a maximum efficiency of 50 percent.

Reece (1980) used an array of flat-plate air-type collectors in combination with a rock thermal storage unit for heating the ventilation air of a broiler production facility. Collector efficiency for the recirculating configuration averaged 50 percent over a 6-day period. Reece found that a system capable of producing 2.9 gigajoules per 1000 broilers during an eight-week production period would provide about 75 percent of the heat required in Mississippi. In addition, Reece recommended 7.74 square meters of flat-plate area per 1000 chickens and 3200 kilograms of rock storage per 1000 birds.

An air-type flat-plate collector was designed by Hall, et. al. (1980) utilizing commonly-available building materials. The collector was operated on a cage-layer house for heating and drying of fecal wastes. Estimates based on feed savings from the supplemental heating showed a five year payback period.

Felton and Brinsfield (1980) compared solar ventilation air preheating with conventional heating using two similar broiler-production test chambers. Both facilities were heated with liquefied-petroleum-gas-fired units. One of the test chambers was fitted with roof-mounted flat-plate solar collectors for preheating the ventilation air. Out of seven test runs, only three resulted in a reduction of liquefied petroleum gas consumption in the solar-assisted unit.

Solar supplemental heating of swine farrowing and growing facilities has been studied by several researchers. Supplemental heat is critical for newborn pigs. As the young pigs grow, the homeothermic mechanisms develop and the need for supplemental heat decreases. However, because of the large quantities of fresh air required for ventilation, the heating demand remains high until the pigs reach a weight of about 18 kilograms (Brewer, et. al., 1981).

Schulte and Veburg (1980) constructed and tested what was termed a "hybrid" solar system on a modified-open-front swine growing facility. The system incorporated both active and passive solar components and included under-floor thermal storage in concrete blocks. The double-pass roof-mounted, active collectors were connected to the block storage in a closed loop. The passive collectors were fiberglass-reinforced panels along the front of the building. The solar components represented an additional investment of fifteen percent of the base cost of the building. With a one percent savings in feed cost, the system was expected to have a payback period of approximately ten years.

DeShazer, et. al. (1980) studied a vertical flat-plate collector attached to the roof of a modified-open-front building. No thermal storage was included. By solar heating of the ventilation air, a reduction of 23 to 24 percent of the needed supplemental energy in Nebraska was achieved.

A vertical wall collector made of concrete blocks and covered with a double glazing of 4-mil Tedlar was constructed and tested by Robbins and Spillman (1980). The concrete acted as both an absorber surface and a storage medium. Material costs were $38.00 per square meter of collector area. The system was constructed against the south wall of a farrowing facility. Air flow rates through the collector ranged from 0.25 to 0.61 cubic meters per minute per square meter of collector, with accompanying efficiencies of 45 percent and 60 percent, respectively. Temperatures of 50 to 60 degrees Celsius were realized on the absorbing surface and tests indicated that an average of 65 percent of the energy was delivered to the building after sunset.

A water-to-air heat pump rated at 12.7 to 19.0 megajoules per hour was used in conjunction with an array of flat-plate, water-cooled collectors for heating a swine nursery in an experiment conducted by Vaughan, Holmes, and Bell (1980). An insulated, covered water pond was used as a storage medium for the solar energy. The COP of the solar-assisted heat pump system was 2.0. The operating cost was competitive with fossil fuel systems and represented an improvement over electric heating.

A 112-square-meter flat-plate collector incorporated into the south wall and roof of a 500-pig nursery was tested by Reeder (1983). The system also included 35 cubic meters of rock thermal storage under the concrete floor. The collector used a recirculating airflow through the rock storage. Ventilation air was preheated as it passed through ducts embedded in the upper layers of the thermal storage. Collector efficiencies ranging from 15 to 40 percent were obtained. Daily temperature peaks of 50 to 60 degrees Celsius in the circulating air were recorded, with an average 25 degrees Celsius drop in air temperature across the storage. The system was estimated to have an eight-to-ten year payback period.

A combination of vertical, wall-mounted, flat-plate collectors and long ducts buried 1.3 meters underground was studied by Kammel and Cramer (1981). The system used solar and geothermal energy to preheat the ventilation air for a 25-stall farrowing house. The solar collectors performed poorly due to problems with controls. Airflow measurements in the underground ducts showed that less air was moving through the building than was expected from the fan ratings. The air temperature at which heat was neither lost nor gained in the underground ducts was about 3.3 degrees Celsius.

Heid and Trotter (1982) noted the potential of solar-heated water for milking parlors. The equivalent of 946 million liters of liquefied petroleum gas was used for milking activities in the U.S. each year. A breakdown of the total energy use showed that water heating accounted for 35 percent. Although the relatively stable year-round demand for dairy hot water is theoretically ideally suited to solar applications (Heid and Williams, 1982), most recent studies have shown that heat exchangers or heat pumps are more economical. Wiersma (1980) compared solar energy and waste heat recovery for milking parlors and concluded that there was no economic reason to choose solar heating over waste heat recovery from refrigeration systems on most dairy farms.

Heid and Trotter (1982) summarized work by USDA researchers on a combination flat-plate collector and waste-heat recovery system for heating water for a 200-cow milking parlor. Payback for the system was from five to ten years, but was highly dependent on the values chosen for particular economic parameters. The study did not include a relative comparison of the heat-exchanger and solar-collector components.

Multi-Use Systems

The potential feasibility for agricultural use of solar energy is greatly enhanced by systems designed for multiple applications (Trotter, Heid, and McElroy, 1979). The longer a system is producing useful output, the lower its fixed cost per unit of energy produced.

Hansen and Smith (1977) investigated the use of a commercially available, double-glazed, flat-plate collector coupled with rock thermal storage for grain drying. An average temperature rise of 8.3 degrees Celsius was obtained using an air flow rate of 1.63 cubic meters per minute per square meter of collector. Additional uses which were planned for the system included home heating and cooling, livestock-building heating and service hot-water heating. Hansen and Smith recommended planning a central solar collector/storage system for new facilities. This would reduce the amount of initial investment in labor and materials. For existing farmsteads, a portable system was thought to be more feasible because the points of application would probably be too scattered for efficient heat distribution from a central location.

Kline (1980) tested and ranked five different configurations of air-type flat-plate collectors for multi-use applications. The collectors were tested with flow rates ranging from 0.61 to 4.27 cubic meters per minute per square meter. The collectors were operated in a double-pass mode for grain drying, and a back-pass-only mode for space and water heating. For grain drying, the noon-hour collector efficiencies were between 44 and 66 percent. Noon-hour efficiencies obtained for water and space heating were between 39 and 50 percent.

Kocher, Bodman, and Lay (1981) reported the performance of a vertical block wall-collector/storage system for preheating the ventilation air of a swine confinement unit and for grain drying in nearby storage bins. The system produced an average temperature rise of 9 degrees Celsius during a mid-January test of ventilation air heating and an average temperature rise of 2 degrees Celsius in the grain-drying mode from mid-October to mid-November. It was estimated that the solar system reduced space-heating energy needs by 39 percent and grain-drying energy requirements in Nebraska were reduced by 4.8 percent.

SDSU SEI-TES System

In 1976, the Agricultural Engineering Department at South Dakota State University began work on a multiple-use solar energy system for agricultural applications. Termed the Solar Energy Intensifier-Thermal Energy Storage (SEI-TES) system, the design was developed with three main components: a flat-plate collector, a parabolic concentrator, and a rock thermal-storage unit (Hellickson, 1980). The system was designed to reduce initial investment cost by utilizing existing air moving equipment in drying bins and livestock shelters, and by increasing the effective receiving area with the reflector component. The reflector could be assembled for a lower cost per unit area than the collector.

The original design, constructed by Julson (1977) and Saienga (1977) centered around a dual-sided, vertical, flat-plate collector, glazed on each side with transparent polyester film (Figure 1). A parabolic reflector made of tempered masonite covered with adhesive-backed aluminum film was focused manually, approximately once a week, on the north side of the collector. Air entered the bottom of the vertical collector on the south side, flowed upward and returned down the north side of the absorber, entering an insulated plenum leading to the application.

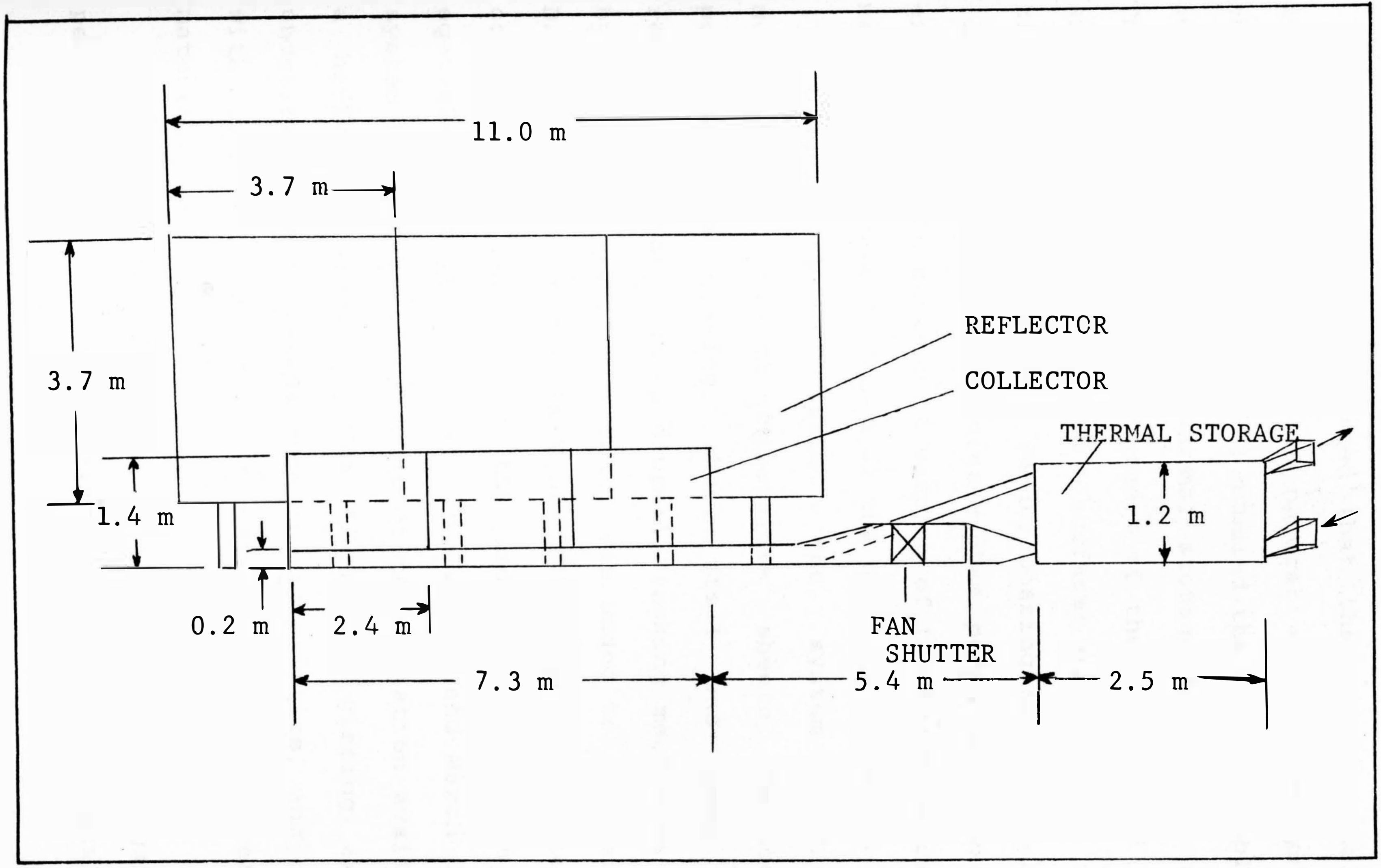

FIGURE 1 . ORIGINAL DESIGN OF THE SEI-TES SYSTEM WITH A DUAL-SIDED, VERTICAL COLLECTOR.

Saienga (1977) reported that the system doubled the drying rate of a comparable natural air drying system for shelled corn. Julson (1977) studied the system coupled with 5000 kilograms of rock thermal storage for space heating. The SEI-TES collected 41 percent of the energy available on an equivalent horizontal surface: a total of 762 kilowatt-hours during the 28-day heating study. Hellickson (1980) noted that the solar-system performance was below expected levels because of warping of the masonite reflector material and deterioration of the polyester glazing.

Siegel (1978) tested the system with major modifications. Clear fiberglass sheets replaced the polyester-film glazing, sheet steel was used as the reflector support, and a diurnal tracking mechanism powered by a 10-watt synchronous motor was added to the reflector. Data collected in the fall of 1977 showed that the system collected 51 percent of the radiation available on an equivalent horizontal area. In January and March 1978,the system collected 36.7 percent of the radiation available on a horizontal surface. The fiberglass glazing exhibited undesirable thermal-expansion characteristics, and problems with wrinkling and peeling developed in the reflective material.

Heber (1979) and Hellevang (1979) investigated the performance of the system with a dual-sided, trapezoidal-shaped collector. The collector was glazed with one layer of low-iron glass. New, polished-aluminum sheets were glued to the reflector. Air entered from the end of the collector and flowed its entire length horizontally in a double-pass pattern. The triangular shape of the collector provided a convenient space for the rock storage during ventilation air preheating.

Hellevang (1979) reported collector average efficiency of 45.6 percent from simulated grain drying with the SEI-TES system. The simulation was achieved by maintaining the system operating parameters at levels normally required for drying shelled corn. Simulation was necessary because the prototype collector, constructed with a wood frame, burned. The frame could not be replaced with metal until after the close of the drying season.

Heber (1979) used 7- to 15-centimeter diameter field rocks for internal thermal storage in the collector. Two separate ventilation-air preheating tests were conducted, each at a different air flow rate. Average efficiency at the lower flow rate was 19.5 percent and at the higher flow rate, 37.5 percent, based on radiation available on the plane of the collector surface. Both Heber and Hellevang found that the long air flow path had an adverse effect on the overall performance of the system.

Astleford (1981) tested the system using a revised air-flow pattern. Air entered the top of the collector, flowed downward between the glazing and the absorber, made a 180 degree turn, and flowed upward between the absorber and a plywood panel acting as a support for the rock storage. The average efficiency of the entire system was 38.9 percent at a flow rate of 0.45 cubic meters per minute per square meter of collector.

Two models using iterative procedures were developed to aid designers in predicting the performance of the SEI-TES system. Polak (1981) developed a computer model of the collector component using basic heat-transfer and fluid-mechanics relationships. This model was intended for use with the system in the grain drying mode. Differences between the model predictions and actual performance data did not exceed 2.5 percent.

Van Zweden, et. al. (1982) used a technique similar to Polak's to adapt the computer model to the ventilation-air preheating mode of operation. Because of varying air-flow rates and the complications induced by the thermal mass of the rocks inside the collector, the results were less accurate than those obtained by Polak (1981). Model predictions were within 2 degrees Celsius of measured output.

The performance of the solar energy intensifier-thermal energy storage system has been investigated under experimental conditions with newly-designed and installed solar units and with careful attention to maintenance of the components for optimal performance. These new units were only field tested over one season.

DESCRIPTION OF RESEARCH FACILITIES

A solar energy intensifier-thermal energy storage system was constructed for preheating the ventilation air of a 12-sow farrowing barn located approximately 5 kilometers west of Sioux Falls, South Dakota (43 degrees N latitude). The building was part of a controlled-access farrow-to-finish operation, and was connected by an alley-way to an 18-sow farrowing unit of identical design, constructed at the same time (1978). Each building was divided into rooms of six sows each (Appendix A, Figure A1). Walls were constructed of aluminum panels on the exterior and interior surfaces with 9 centimeters of fiberglass batt insulation between. The ceiling and roof were also covered with aluminum paneling. The ceiling was insulated with 25.4 centimeters of blown cellulose. Floors were concrete with 5-centimeter-thick expanded polystyrene perimeter insulation extending 0.6 meters below grade along the interior of the foundation. The floor near the front of each farrowing crate was heated with hot water through polyvinyl-chloride pipe embedded in the concrete. A 2.4 meter wide by 1.2 meter deep manure pit ran the length of each building. The calculated average thermal conductance of the buildings was 0.29 watts per square meter per degree Celsius.

Each room in the farrowing section had a negative pressure ventilation system consisting of a thermostatically-controlled, variable-speed fan (3 to 40 cubic meters per minute) for winter ventilation, and a constant-speed fan (100 cubic meters per minute) for summer ventilation (Appendix A, Figure A1).

In winter, fresh air was drawn through an inlet under the eave on the south side of the building, across the ceiling through the attic and into the rooms through a long, baffled, 0.15 meter wide slot with an adjustable opening located in the north wall near the ceiling. In summer, a hinged trap-door 0.15 meter-wide along the entire length of the building under the north eave was dropped open to allow fresh air to enter the rooms directly from the north side of the building. In addition to the two fans, each 6-sow room was equipped with a propane-fired heater for supplemental heating.

A plan view of the farrowing unit with the solar system is shown in Figure 2. The system was constructed to the north of the 12-sow building. The 12-sow barn was the solar-assisted facility. The 18-sow unit remained a conventionally-operated structure for comparison purposes.

The supplemental solar heating system, installed in the fall of 1980, consisted of a 3.0 x 7.3-meter parabolic reflector/concentrator, and a 0.76 x 4.88-meter trapezoidal

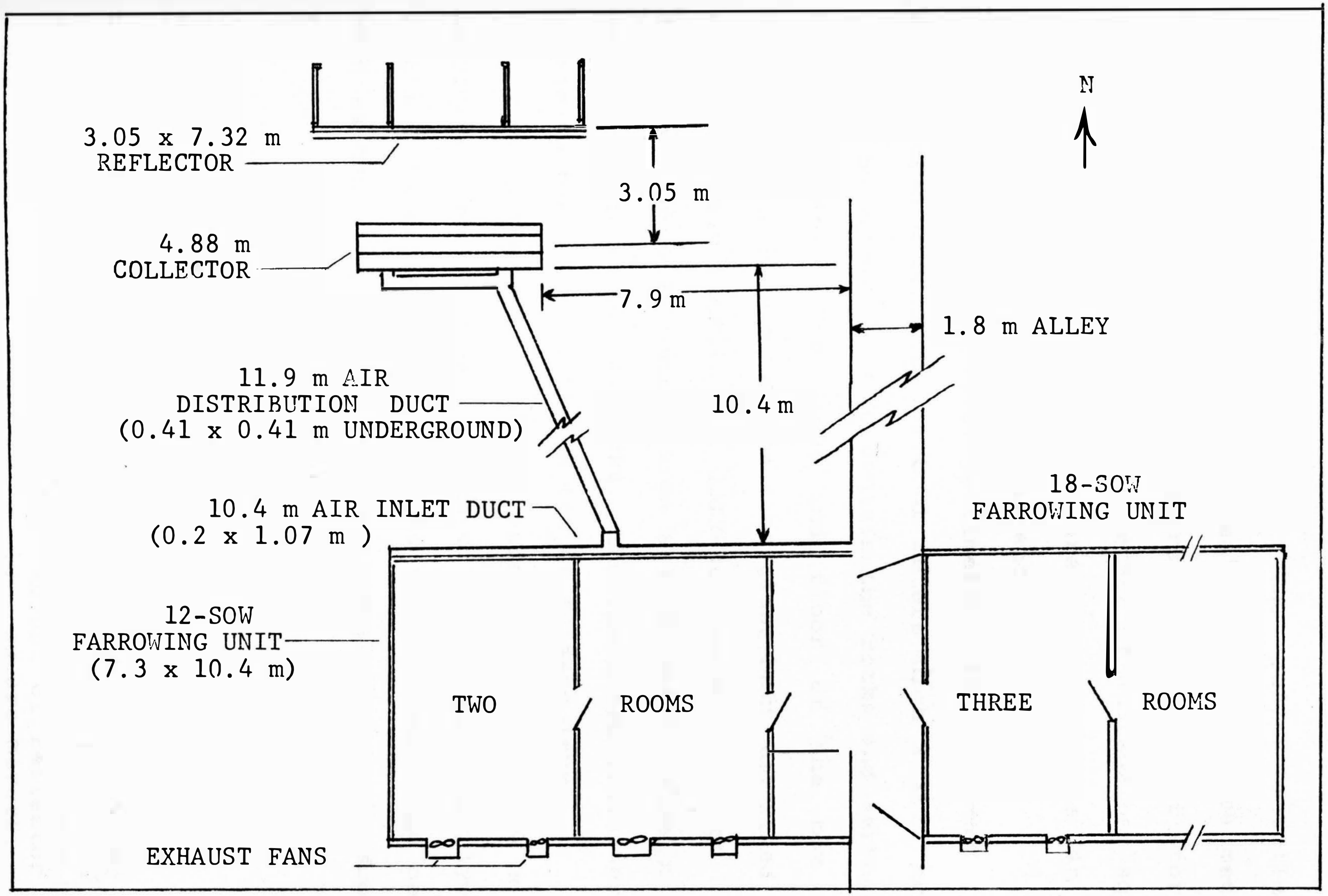

FIGURE 2 . PLAN VIEW OF FARROWING UNITS AND SUPPLEMENTAL SOLAR HEATING SYSTEM.

shaped, dual-sided collector. The reflector-collector combination had an effective area[1] of 19.1 square meters. The collector housed 4 cubic meters of rock thermal storage, providing a ratio of 0.21 cubic meters of storage per square meter of effective collector area. Void space in the storage was approximately 40 percent.

Exterior plywood, 1.27 centimeters thick, formed a box with outside dimensions of 0.61 meters high x 1.22 meters wide x 4.88 meters long to contain the rocks and serve as a collector base. The sides and floor of the box were insulated with 0.15 meters of fiberglass batt and lined with an inner box built of 1.27-centimeter plywood with dimensions of 0.46 meters high x 0.92 meters wide x 4.57 meters long to store the rocks. The insulated container was placed in the ground with the top at surface level.

The collector frame was made of 0.32 x 1.9 x 1.9 centimeter steel angles. Sheets of 1.27-centimeter plywood, 0.76 x 2.44 meters, were attached to the frame, and served both as one side of the airflow channel, and as a support for the rocks inside the collector.

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[1] "Effective area" is defined as the intercepted solar area and includes the 4.2 square meter south side of the collector and 14.9 square meters of the 22.3 square meter reflector (which is the maximum amount of reflector area used at a time), for a total of 19.1 square meters.

The dual-sided collector was 4.88 meters long x 0.76 meters high. Each side was composed of two 0.86 x 2.44 meter sections at an angle of 60 degrees from the horizontal. Each section consisted of a single 4.88-millimeter, plate of low-iron glass glazing (approximate transmissivity of 0.9) over a 1.25-millimeter sheet-metal absorber painted flat black (absorptivity and emissivity about 0.9). A sheet-metal cover, insulated with 2.54 centimeters of styrofoam sealed the storage area and protected the collector inlets.

A layer of concrete blocks, spaced 2.54 centimeters apart in the bottom of the rock storage box, formed a low-resistance air-flow passage to enhance even air distribution through the rocks to the system outlet. Field rocks averaging 15 centimeters in diameter were used as the thermal storage medium.

The air-flow pattern in the system, depicted in Figure 3, was double-pass through the collector and single-pass through the storage. Air entered the inlet at the top of the collector, flowed downward between the glazing and the absorber, then made a 180 degree turn and flowed upward between the absorber and the plywood rock support. At the top of the collector, air entered the storage system and flowed downward through the rocks. At the bottom of the thermal storage area, the air flowed through an outlet into

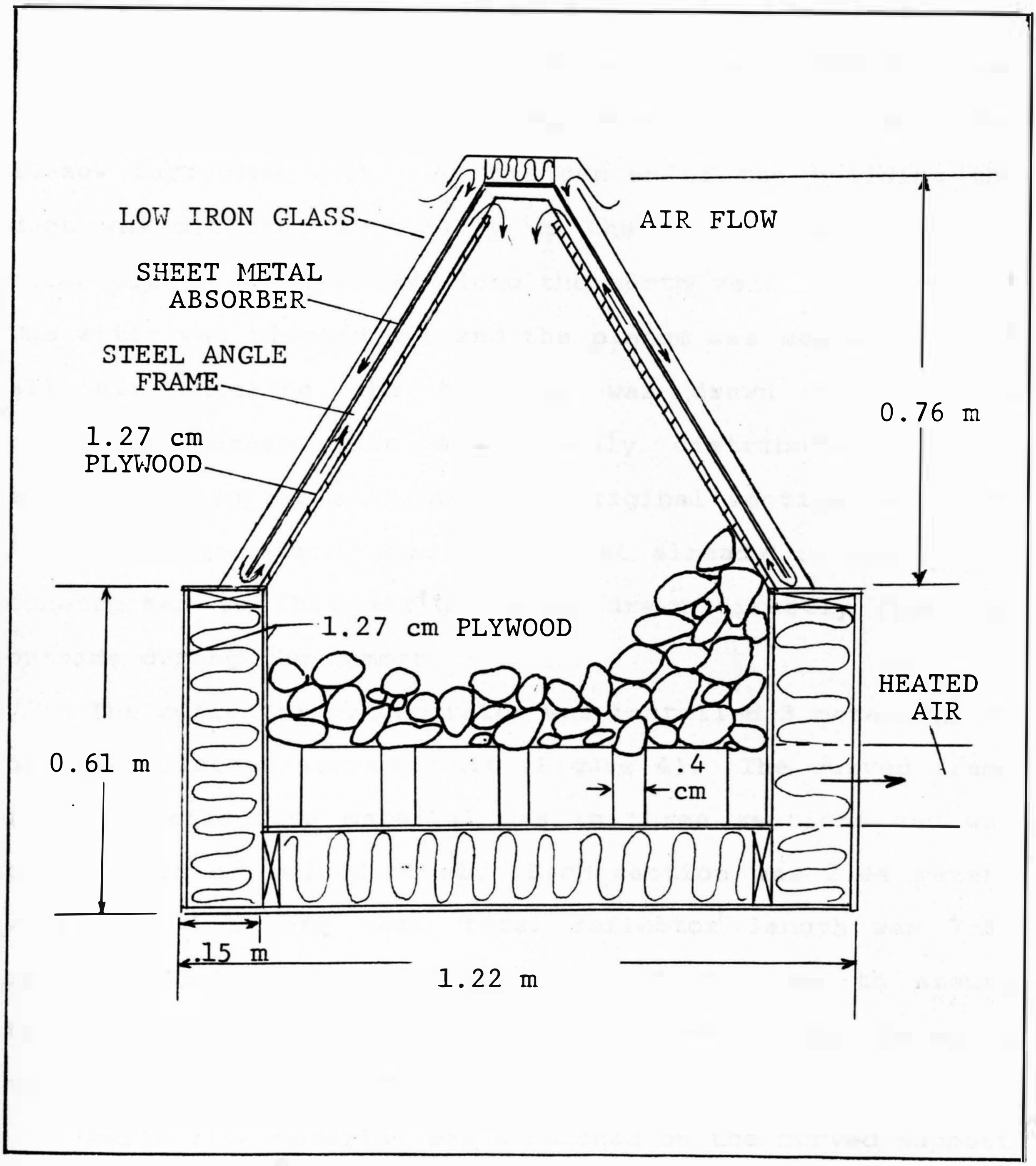

FIGURE 3 . CROSS-SECTION OF THE SCLAR COLLECTOR-THERMAL ENERGY STORAGE UNIT.

a 0.4 x 0.4 meter duct built of 1.27-centimeter plywood and insulated with 3.8 centimeters of styrofoam. The duct was buried 0.3 meters underground and extended 12 meters to the 12-sow farrowing unit. At the center of the building the duct was directed vertically up the wall to a 0.2 x 1.1 meter plenum constructed along the north ventilation inlet. The attic was blocked off and the plenum was sealed so that all air entering the building was drawn through the collector/storage unit and finally distributed in the solar-assisted rooms through the original ventilation slot. A long, hinged door, similar to that already in place, was constructed so that air could be drawn directly from the outside during the summer.

The reflector/concentrator was installed 3 meters north of the collector/storage unit (Figure 4). The curved frame for the reflective material was in three sections and was constructed of welded steel. Each section was 2.44 meters wide x 3.05 meters high; total reflector length was 7.32 meters. The extra length of the reflector was to assure that the collector area was fully utilized during the early morning and late afternoon hours.

Reflective material was stretched on the curved support and held fast by adjustable clamps at each end of the frame. During the 1981-82 season, YS-91, (manufacturer-rated reflectivity of 0.85), a metalized polyester film was used

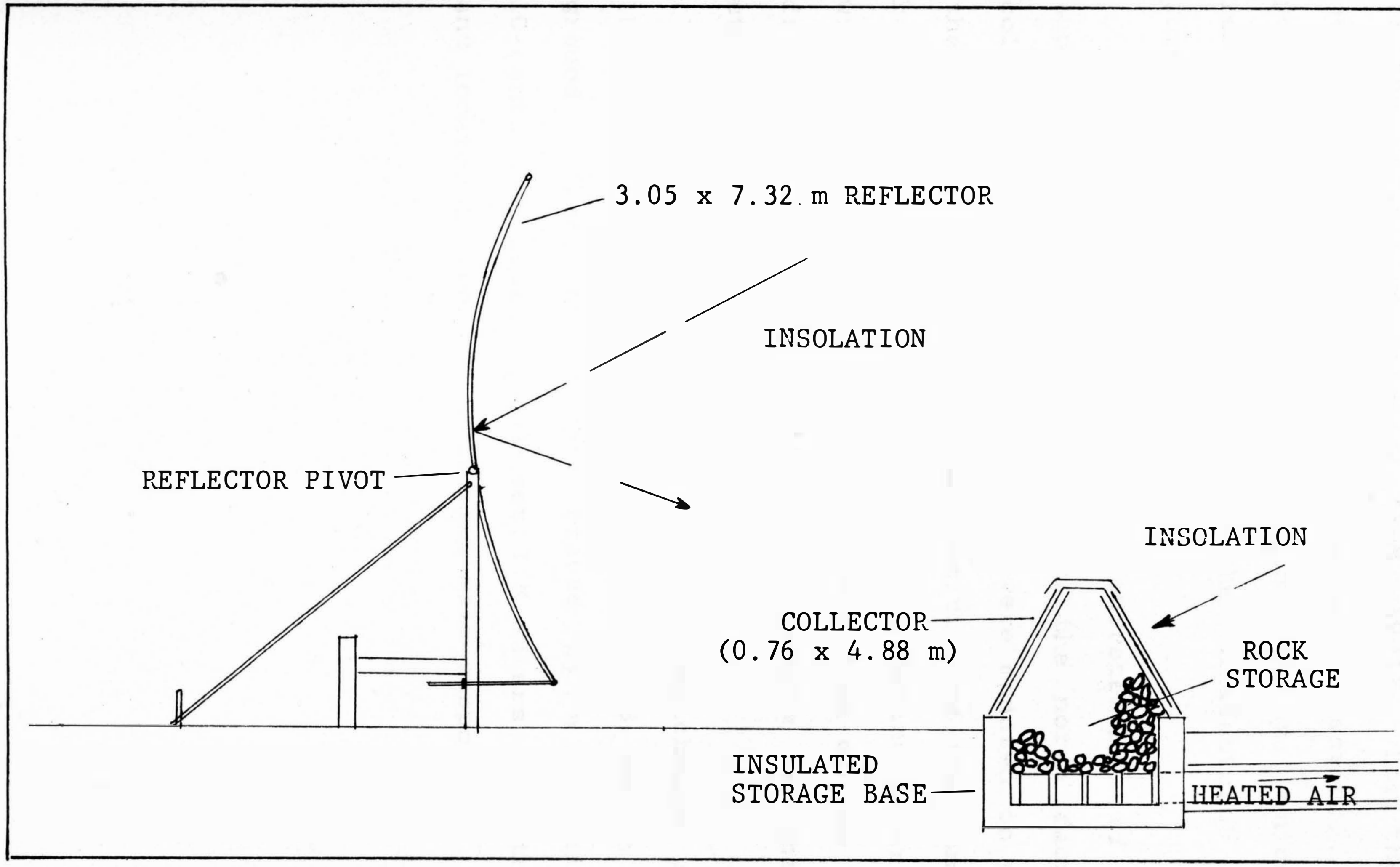

FIGURE 4. SOLAR ENERGY INTENSIFIER-THERMAL ENERGY STORAGE SYSTEM FOR PROVIDING SUPPLEMENTAL HEAT TO THE 12-SOW FARROWING UNIT.

as the reflective material (3-M, 1981). For the 1982-83 season, the YS-91 was replaced with heavier, flexible, polished aluminum sheets called Kinglux, which had a reflectivity of 0.9 rated by the manufacturer (Kingston Corp., 1981)

The reflector produced a concentrated band of radiation approximately 0.5 meters wide on the north face of the collector. The reflector frames were mounted on pivots on the support structure and the position of the concentrated band was adjusted for the seasonally varying solar altitude with threaded rods attached to the bottom of the reflector frame, running through threaded brackets on the support structure.

The reflector support structure was channel steel and 10-centimeter-square wooden posts set 1.8 meters into the ground. These posts were braced by a second set of 10-centimeter-square posts set 1.8 meters into the ground and located 1.2 meters behind the main supports.

TESTING AND ANALYSIS PROCEDURE

Instrumentation

Four basic measurements were essential to evaluate the performance of the SEI-TES system and compare it with the conventionally heated farrowing unit. These measurements were: insolation, temperatures, air flow rates, and fuel consumption.

A microprocessor-controlled data-acquisition system recorded and processed data. Data were measured hourly, and simultaneously printed on hardcopy and recorded on tape.

Insolation was measured with a pyranometer mounted at a 60 degree angle, 1.83 meters above ground near the centerline of the reflector. Instantaneous insolation was measured at a rate of 5 readings per second continuously over a 5-minute period at the beginning of each hour. The 5-minute average was used as the hourly radiation intensity.

Temperatures were monitored with type T copper-constantan thermocouples. Temperature profiles were recorded in the collector and rock storage in each of the two 2.44-meter sections. Temperatures at the outlet of the rock storage, at the inlet to the building, and inside each of the farrowing units were also recorded. These temperatures as well as the outside air temperature were measured hourly.

Air flow was measured with a temperature-compensated hot-wire anemometer. Velocity profiles were recorded in a planar cross-section perpendicular to the flow in the duct. Flow rates were calculated and a point was located which, when multiplied by cross-sectional area, yielded flow rate. The anemometer probe was placed at this point for long-term testing. Duct velocity readings were recorded hourly.

Propane use in the farrowing units was monitored with two positive displacement gas meters in the fuel lines. The first meter was placed in the line at the point of entry to the farrowing section of the swine complex. This meter indicated total fuel flow into the solar and conventional units. The second meter measured fuel flow into the solar heated rooms. By subtracting the reading on the second meter from the total on the first, flow of propane into the conventionally heated unit was calculated. Meter readings were recorded approximately once per week by personnel at the facility. Since the meters were calibrated in hundreds of cubic feet, readings did not often change from one day to the next, especially in mild weather. Because of this, fuel consumption comparisons were only valid over periods of at least one week.

During the 1981-82 heating season the measured flow rate through the solar system ranged from 0.029 to 0.084 cubic meters per second. This was lower than the recommended minimum winter ventilation rate of 0.12 cubic meters per second for the 12-sow farrowing unit. It was thought that the anemometer was out of calibration, so it was recalibrated.

When data collection resumed during the 1982-83 heating season, the same problem was encountered. Static pressure and fan speed measurements in the building revealed flow rates of 0.35 to 0.40 cubic meters per second. Investigation revealed that the discrepancy was due to infiltration. Tests with smoke candles exposed leaks around the plenum on the north side of the farrowing unit. These leaks allowed a substantial volume of air to bypass the solar system and enter the building directly. The cracks, which were not evident during visual inspection, were apparently due to normal shrinkage and weathering of the duct materials.

The SEI-TES system was designed to draw all of the ventilation air through the collector/storage component. Because of the significant infiltration from other sources, this did not occur. The result was lower solar system efficiencies than would be expected with higher air flow rates.

Data Analysis

Three important system parameters were chosen as dependent variables.[2] These were collector efficiency, system efficiency, and total energy output of the system. They are calculated as follows:

$$\text{COLL} = mc_p\ (\text{Tcol} - \text{Tamb})/\Sigma\,\text{Imeasured}$$

$$\text{SYS} = mc_p\ (\text{Tsys} - \text{Tamb})/\Sigma\,\text{Imeasured}$$

$$\text{TOTAL} = mc_p\ (\text{Tsys} - \text{Tamb})$$

Of the thirty-four quantities measured, three were selected by step-wise linear regression as significant independent variables. The independent variables were: available insolation, air-flow rate, and ambient temperature.

Because of the transient nature of the thermal energy being collected, stored, and removed for heating, a "day" was defined as a 24-hour period beginning at 0800 and ending at 0800 the following morning. During this time period the system normally received, stored, and dissipated the quantity of energy available as incoming solar radiation for the given day. Hourly readings for insolation and total energy delivered were summed over the 24-hour period. System efficiency, ambient temperature, and air flow rate were averaged for the 24-hour period. Collector efficiency

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[2] Parameter and variable abbreviations for the "Testing and Analysis Procedure" and "Results" sections are listed in Table 1.

Table 1 List of Symbols.

Variable	Description
AIRF	Air flow rate through the system, m^3/sec
COLL	Collector efficiency, decimal percentage
c_p	Specific heat of air at constant pressure, $kJ/kg \cdot {}^oC$
INSOL	Daily total available insolation, kWh
$\sum I_{measured}$	Sum of insolation available on a 60^o surface for the day, kWh
m	Mass flow rate of air, kg/s
SYS	System efficiency, decimal percentage
T_{amb}	Ambient temperature, ^oC
T_{col}	Collector output temperature, ^oC
T_{sys}	System output temperature, ^oC
TOTAL	Daily total energy output for the system, kWh

was averaged only over the hours when insolation was available.

To evaluate the performance of the system and develop a useful relationship for predicting energy output, 41 days of data from two consecutive heating seasons were analyzed. A step-wise multiple linear regression was performed using average ambient temperature, average daily air flow rate, and total daily insolation to predict collector efficiency, system efficiency, and total energy delivered on a daily basis.

The system was dismantled after testing was completed and the components were checked for wear and deterioration.

RESULTS

The results of the project include analyses of collector efficiency and system performance, a comparison of conventional energy use in the solar-assisted and control units, and a summary of the economic implications of the performance data. In addition, a section devoted to system reliability describes the effects of weathering and deterioration on the collector and reflector components.

Collector Efficiency

The design of the collector/storage system was effective in minimizing heat losses without insulating the upper portion of the rock storage chamber. This is evident by noting that the collector output continued to exhibit a temperature rise several hours after insolation was no longer available (Figure 5). A small portion of this temperature rise may be attributed to transient heat in the collector materials, but the major component of the temperature rise is due to heat convected back into the system after being conducted from the storage area through the plywood and into the air channel. This extended temperature rise was noted in previous studies of the SEI-TES system.

Heat transfer back into the collector air channels from the rock storage made evaluation of the collector efficiency

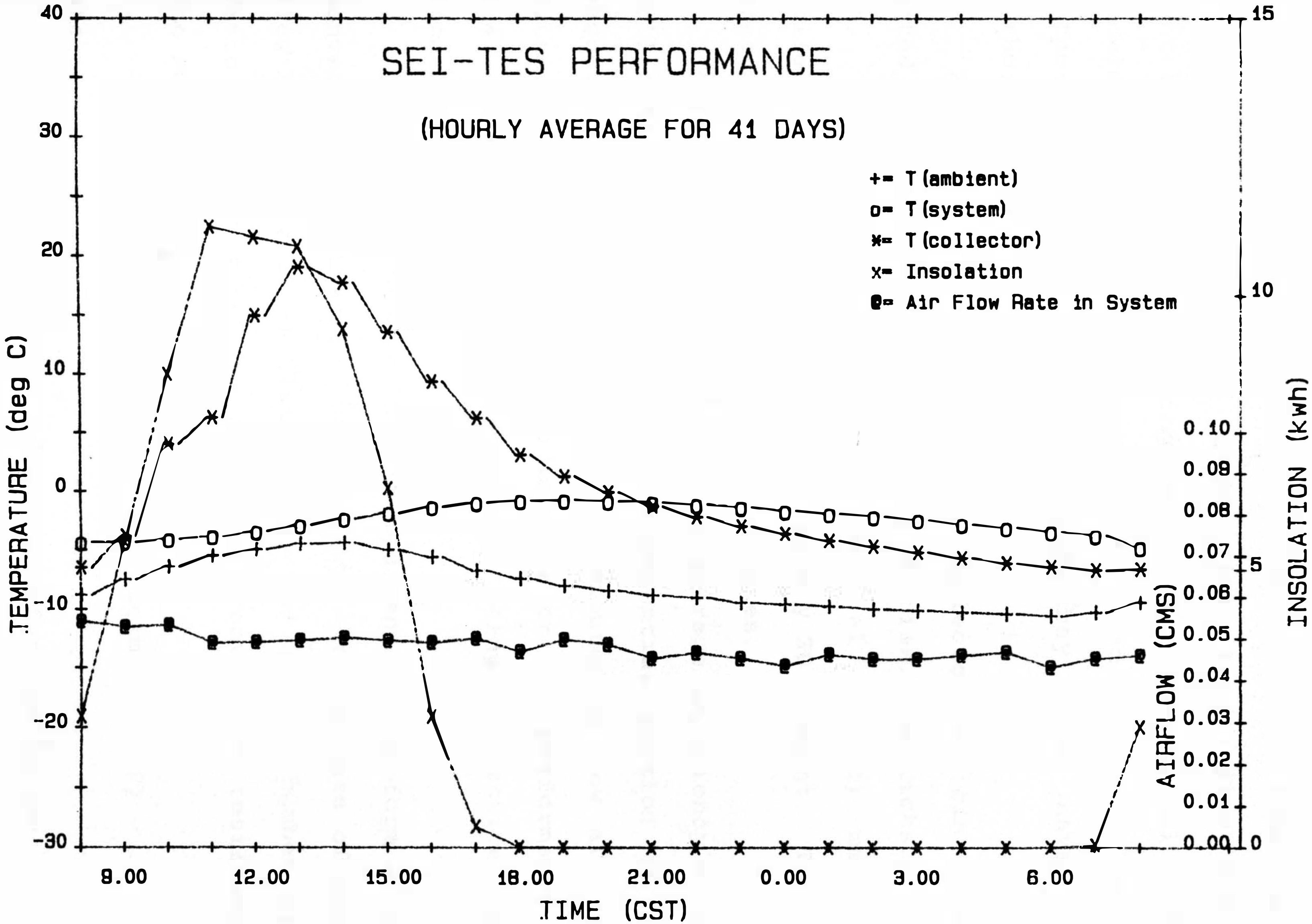


FIGURE 5. AVERAGE HOURLY SEI-TES SYSTEM PERFORMANCE.

difficult. Therefore, conclusions about the performance of the collector component drawn from data taken during ventilation air preheating must be tempered with the knowledge that the temperatures measured at the collector output were probably higher than they would have been without the rock storage.

A significant reduction in collector efficiency was noted when compared with previous studies. Hellickson, et. al. (1981a) and Hellickson, et. al. (1982) measured collector efficiencies of approximately 30 percent. Table 2 presents the results of previous studies.

During the 1981-83 study, the average efficiency of the collector was 18 percent. A substantial portion of the reduction in efficiency can be attributed to low air flow rates. The effect of air flow rate on the performance of the collector can be quantified using convective heat transfer principles.

The Nusselt number, Nu, is a dimensionless form of the convective heat transfer coefficient. In the case of fluid flow between parallel flat plates, the Nusselt number is a ratio of conductive resistance to convective resistance. The Nusselt number is defined as

$$Nu = h_c L / k_f \qquad \text{(Duffie and Beckman, 1980)}$$

where

h_c =the convective heat transfer coefficient,

L= a characteristic length, and

Table 2 Flowrates and efficiencies measured in previous studies of ventilation air preheating with the SEI-TES system.

Investigators	Flow Rate	System Efficiency	Collector Efficiency
Hellickson, Christianson, and Heber, 1981	0.005 $m^3/s \cdot m^2$	24.5%	31.2%
Van Zweden, Hellickson, and Christianson, 1982	0.002 - 0.010 $m^3/s \cdot m^2$ (variable)	24.0 %	26.4%

k_f =conductivity of the working fluid.

The Nusselt number is related to two other dimensionless parameters, the Reynolds number, Re and the Prandtl number, Pr. The relationship is of the form

$$Nu= (C)(Re^{m})(Pr^{n})$$ (Kreith and Kreider, 1978)

where C, m, and n are determined from experimental data to form an empirical equation.

The Reynolds number, a dimensionless ratio of inertial forces to viscous forces, can be calculated for a specific situation by

$$Re= (\rho)(v)(Dh)/(\mu)$$ (Kreith and Kreider, 1978)

where

ρ = density of the fluid,

v= average velocity,

Dh= hydraulic diameter, and

μ = dynamic viscosity. The hydraulic diameter for two parallel flat plates is defined by Duffie and Beckman (1980) as twice the plate spacing.

The Prandtl number, a ratio of momentum diffusivity to thermal diffusivity, is defined by

$$Pr= c_p \mu /k_f$$ (Kreith and Kreider, 1978)

where

c_p = specific heat of the fluid,

μ = dynamic viscosity, and

k_f = conductivity of the fluid.

Empirical relationships for the Nusselt number for air flow between parallel flat plates with heating on one side are numerous. Two of the most commonly cited equations are: for laminar flow:

$$Nu = 4.9 + \frac{0.606(RePrDh/L)^{1.2}}{1 + 0.0909(RePrDh/L)^{0.7}\,Pr^{0.17}}$$

(Duffie and Beckman, 1980)

for turbulent flow:

$Nu = 0.023Re^{0.8}\,Pr^{0.4}$ (Kreith and Kreider, 1978).

These two relationships were used for calculating the Nusselt number associated with the SEI-TES collector. An average air temperature of 290 degrees Kelvin was assumed. In addition it was assumed that there was a 3:1 mass flow ratio for air entering the north and south sides of the collector.

For the average flow rate of 0.0025 cubic meters per second per square meter of effective collector area measured during the 1981-83 study, the Reynolds number ranged from 300 to 1000, indicating that the flow was definitely in the laminar regime. Using the relationship for the Nusselt number for laminar flow, an average convective heat transfer coefficient of 9.2 watts per square meter per degree Celsius was calculated. The convective heat transfer coefficient would have been 2.5 times larger if the system had operated as it was designed. The flow rate would have been approximately 0.018 cubic meters per square meter of effective area, producing Reynolds numbers in the turbulent range. The heat transfer coefficient, calculated from the turbulent relationship for the Nusselt number, would have averaged about 23.6 watts per square meter per degree Celsius. The reduction in heat transfer rate from the absorber to the airstream seriously affected the performance of the collector.

Figure 6 presents a plot of collector efficiency versus air flow rate. Regression analysis developed a linear relationship which was significant at the 0.05 level, however the r-squared correlation coefficient was only 0.17:

$$COLL = 6.441 + 235.326(AIRF).$$

Weathering and deterioration, described in the System Reliability section, affected the collector efficiency by reducing the transmissivity of the glazing and the absorptivity of the plate. It was not possible to accurately quantify the effects of reduced absorptivity and and transmissivity.

Another factor which reduced the collector effectiveness was snow and ice accumulations in and around the collector/storage unit. Two severe winter storms occurred during the 1981-82 season, one on 9 January 1982,

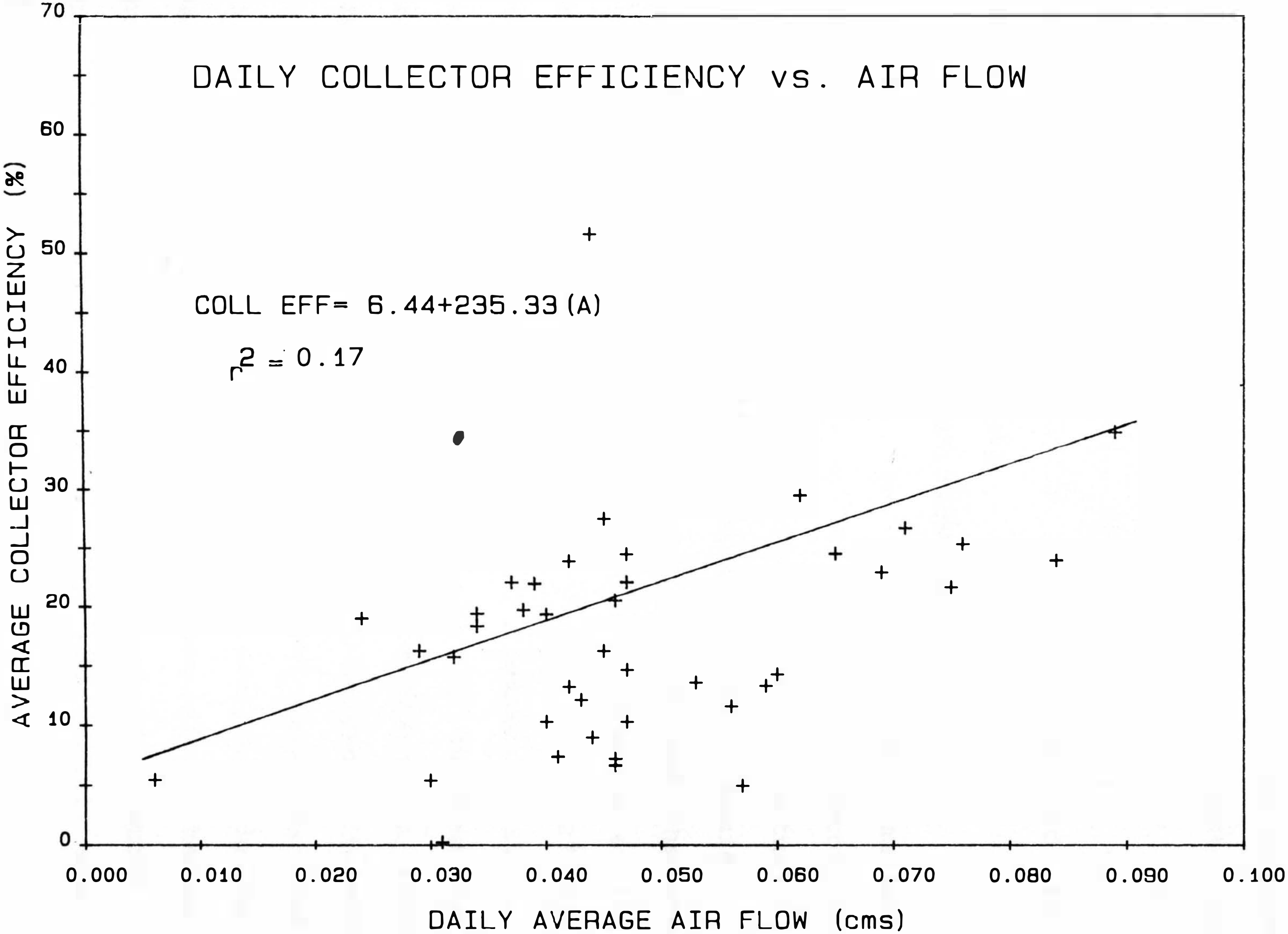


FIGURE 6. DAILY AVERAGE EFFICIENCY OF THE COLLECTOR vs. AVERAGE AIR FLOW RATE.

and the other on 25 January 1982. After each of these storms, the collector/storage unit was found completely covered with snow. Strong northwesterly winds were observed depositing drifting snow between the glazing and the absorber on several occasions. On days of very low insolation this accumulation of snow remained in the collector throughout the day.

System Performance

System output was the most accurate method of evaluating the performance of the entire collector/storage system. For purposes of evaluation under actual field conditions, analyses were conducted on a daily basis.

Table 3 summarizes the system design and the average performance characteristics. The system average efficiency of 13.8 percent was well below the 24.5 to 27 percent average noted in the previous studies. Since the performance of the system was directly related to the performance of the collector, the problems which affected the collector also affected the overall system efficiency.

Energy provided during the 41 test days averaged 9.0 kilowatt-hours per day. Insolation available during the testing averaged 68.1 kilowatt-hours per day. The maximum energy provided by the system occurred on 28 December 1981, and was 34.9 kilowatt-hours when the total available radiation was 121.1 kilowatt-hours. The minimum energy

Table 3 SEI-TES Performance for farrowing unit ventilation air heating.

Parameter	Average	Total
Number of Agricultural Units		12 sows & litters
Collector Area	1.59 m^2/A.U.*	19.1 m^2
Collector Flow Rate	0.0025 $m^3/s \cdot m^2$	
Average System Temperature Rise	5.2° C	
Average Ambient Temperature	-7.8° C	
Average Insolation Available	68.1 kWh/day	
Energy Provided (41 days)	9.0 kWh/day	369 kWh
Propane Energy Used (98 days)	63.78 kWh/day (8.9 l/day)	6251 kWh (873 l)
Average Efficiency		
Collector	17.7%	
System	13.8%	

* A.U. = Animal Unit (1 sow and litter).

provided was -3.4 kilowatt-hours on 20 December 1981 when the total radiation available was 43.1 kilowatt-hours. This net loss of energy was recorded due to a shift in the charge/discharge cycle of the thermal storge unit. The preceding day was characterized by very low insolation levels and subnormal ambient temperatures. The thermal storage was brought to a low temperature of approximately -12 degrees Celsius. In the early morning hours of 20 December the ambient temperature began to rise steadily and reached a peak of near 0 degrees Celsius by 1200 hours. A low insolation level was also measured on 20 December. Because of the low temperature of the thermal storage, there was a period of approximately 17 hours during which the system output temperature was lower than the ambient temperature.

Average daily temperature rise of the system was 5.2 degrees Celsius. The average hourly temperature curve (Figure 6) showed an average of 7 hours time lag between the maximum collector temperature and the maximum system output temperature. Collector temperature peaked at 1300 hours on days with moderate to high insolation levels (greater than 45 kilowatt-hours per day). System temperature usually peaked at 1900 or 2000 hours on those days. The system output temperature normally remained within 3 degrees Celsius of maximum for about 3 hours.

For days with less than 45 kilowatt-hours of total radiation, the temperature output followed a similar pattern with peak output influenced mainly by the ambient temperature rather than insolation.

Figures 7 through 13 illustrate a typical week of operation of the system. The effects of insolation level and ambient temperature on the system output are shown. Figure 14 represents the cumulative totals of energy available and system output during the study.

Step-wise multiple regression produced a highly significant relationship for predicting system output using the variables insolation, air flow, and ambient temperature. The equation was:

$$TOTAL = -11.865 + 249.996 AIRF - 0.395 T_{amb} + 0.086 INSOL$$

The correlation coefficient for this relationship was 0.76 and the standard error of estimate was 0.02.

Figures 15, 16, and 17 present plots of system output versus insolation, airflow, and ambient temperature respectively. The regression line for each individual relationship has been super-imposed on the plot. Hourly energy received, collected, and delivered by the system for an average day are depicted in Figure 18.

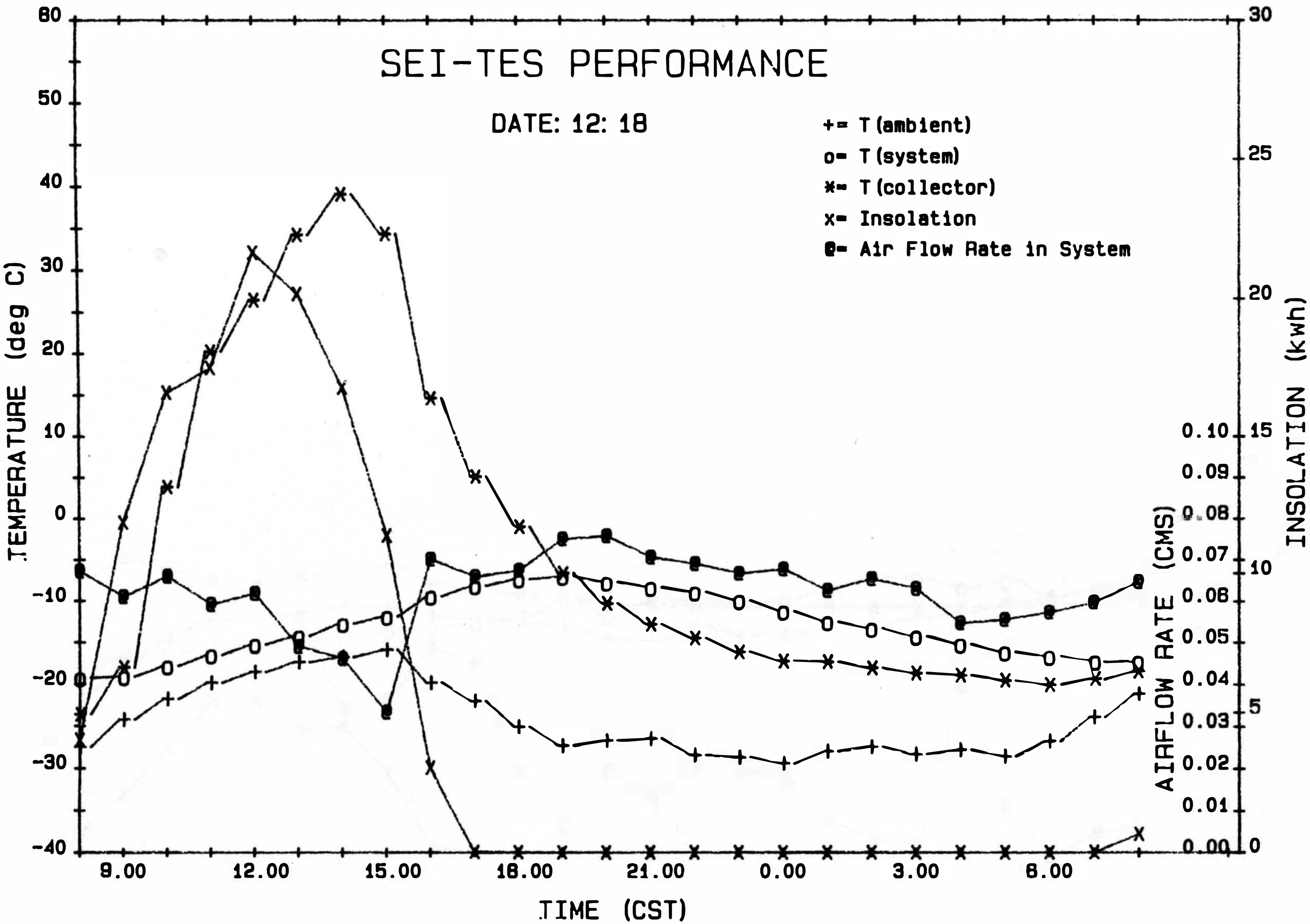


FIGURE 7 . HOURLY SEI-TES SYSTEM PERFORMANCE, 18 DECEMBER, 1981.

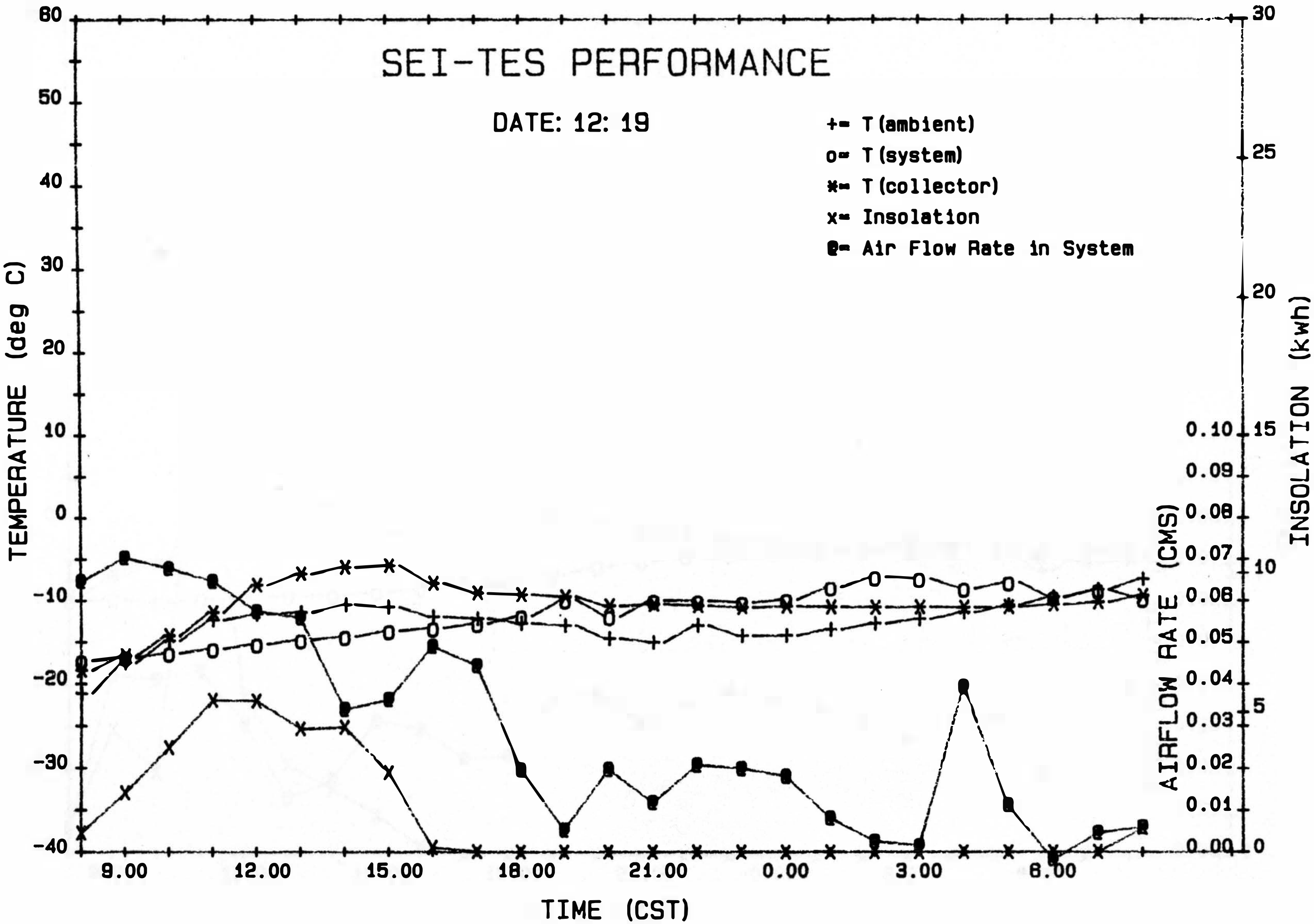
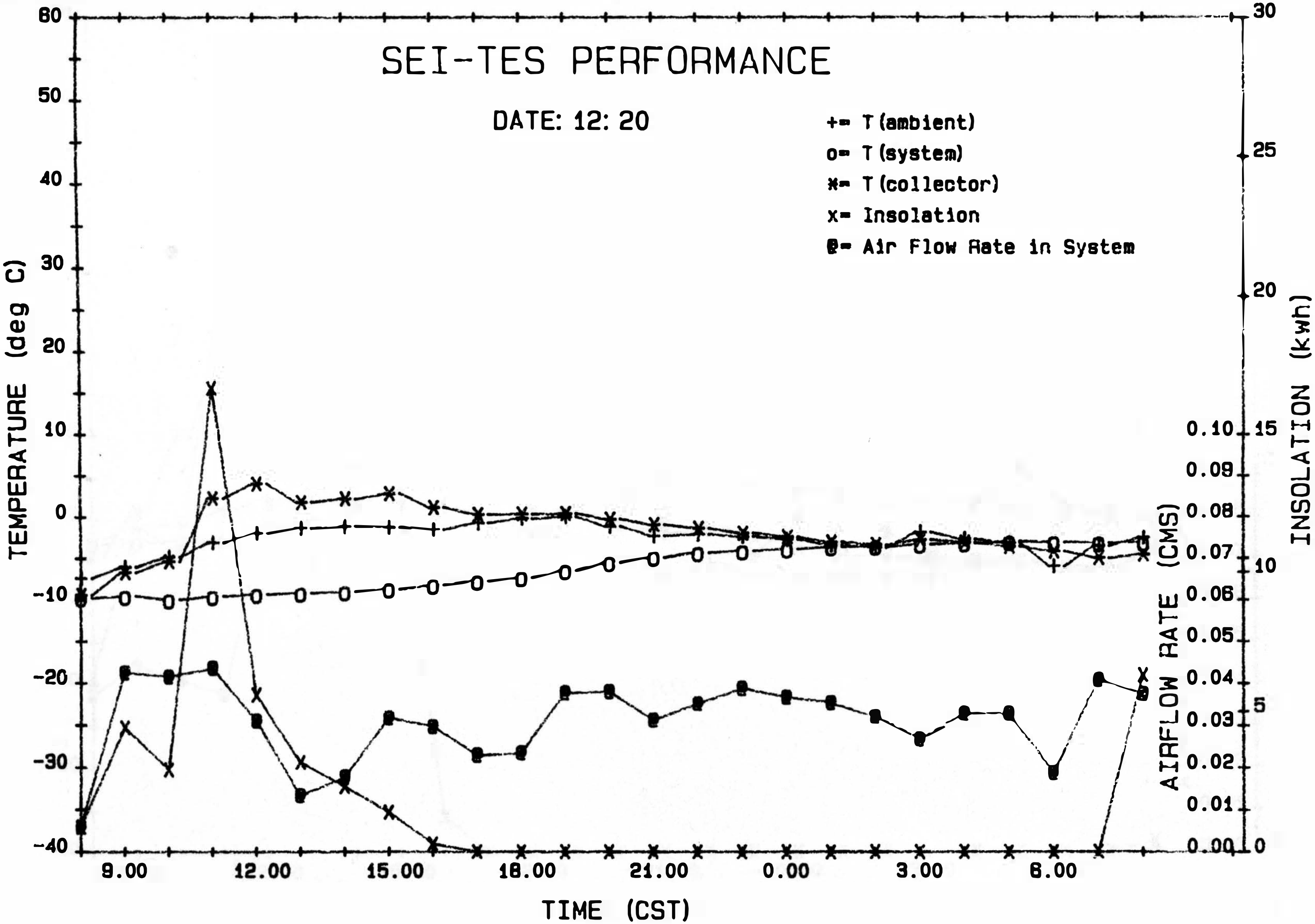


FIGURE 8 . HOURLY SEI-TES SYSTEM PERFORMANCE, 19 DECEMBER, 1981.



.FIGURE 9. HOURLY SEI-TES SYSTEM PERFORMANCE, 20 DECEMBER, 1981.

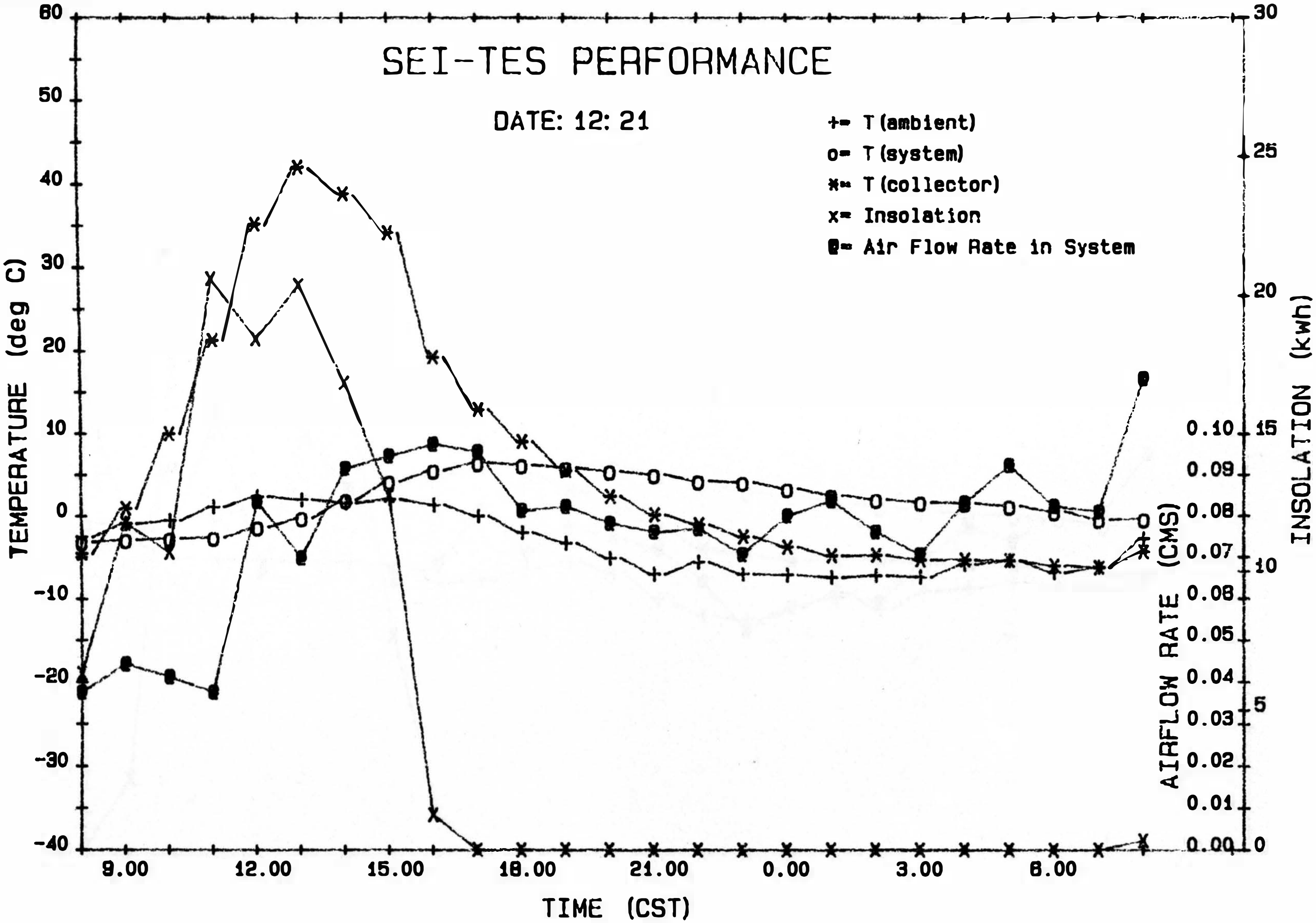


FIGURE 10. HOURLY SEI-TES SYSTEM PERFORMANCE, 21 DECEMBER, 1981.

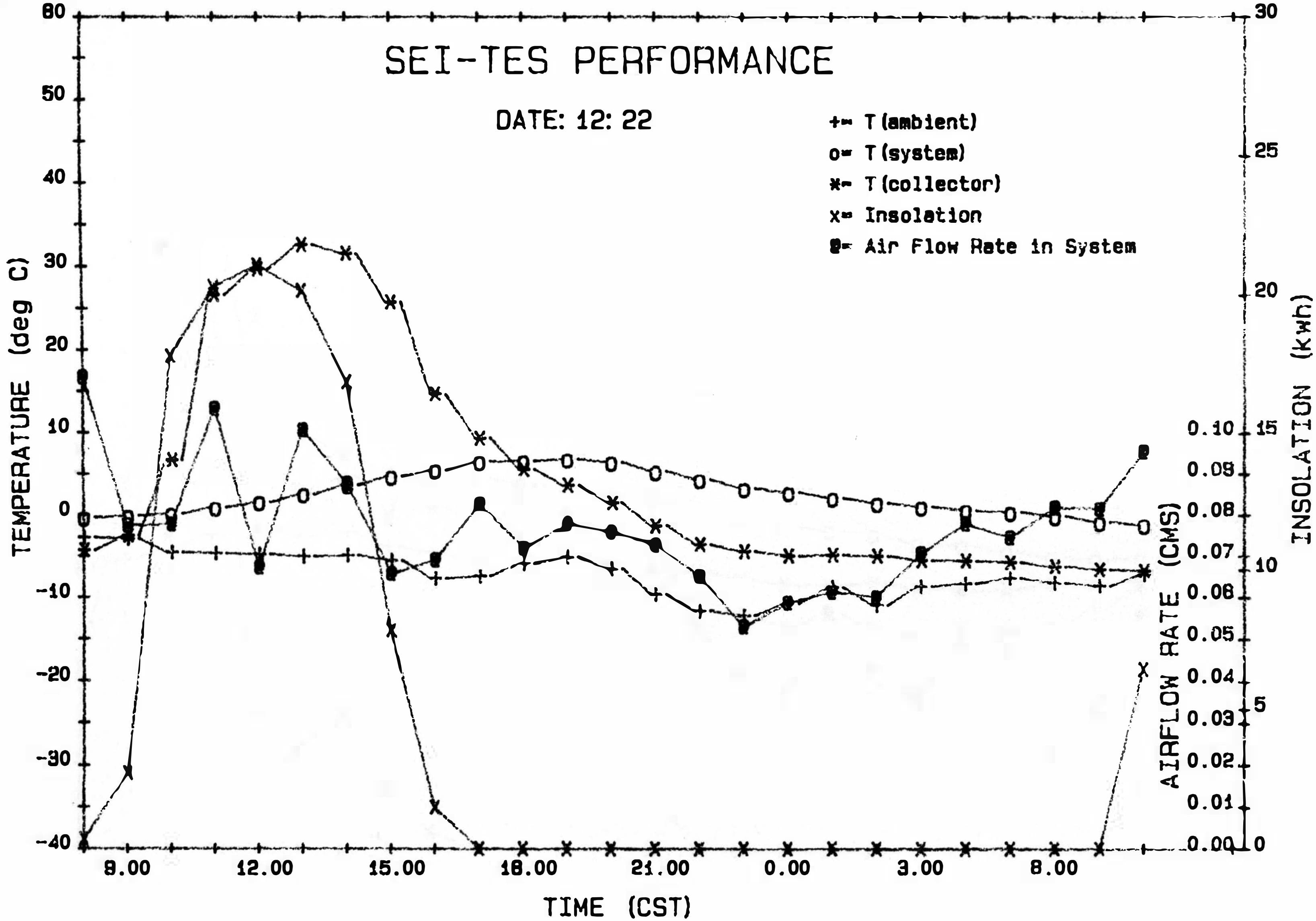


FIGURE 11. HOURLY SEI-TES SYSTEM PERFORMANCE, 22 DECEMBER, 1981.

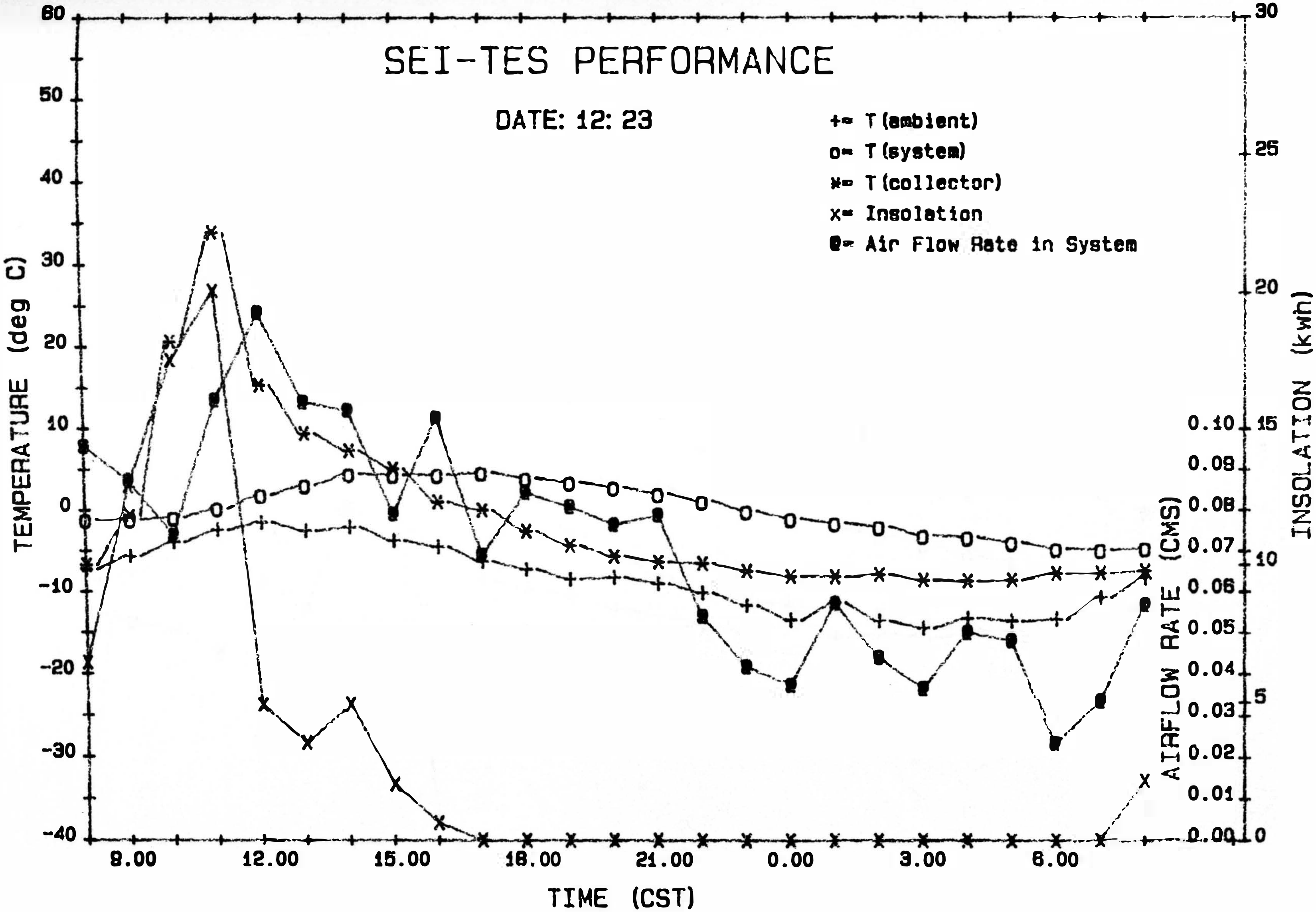


FIGURE 12. HOURLY SEI-TES SYSTEM PERFORMANCE, 23 DECEMBER, 1981.

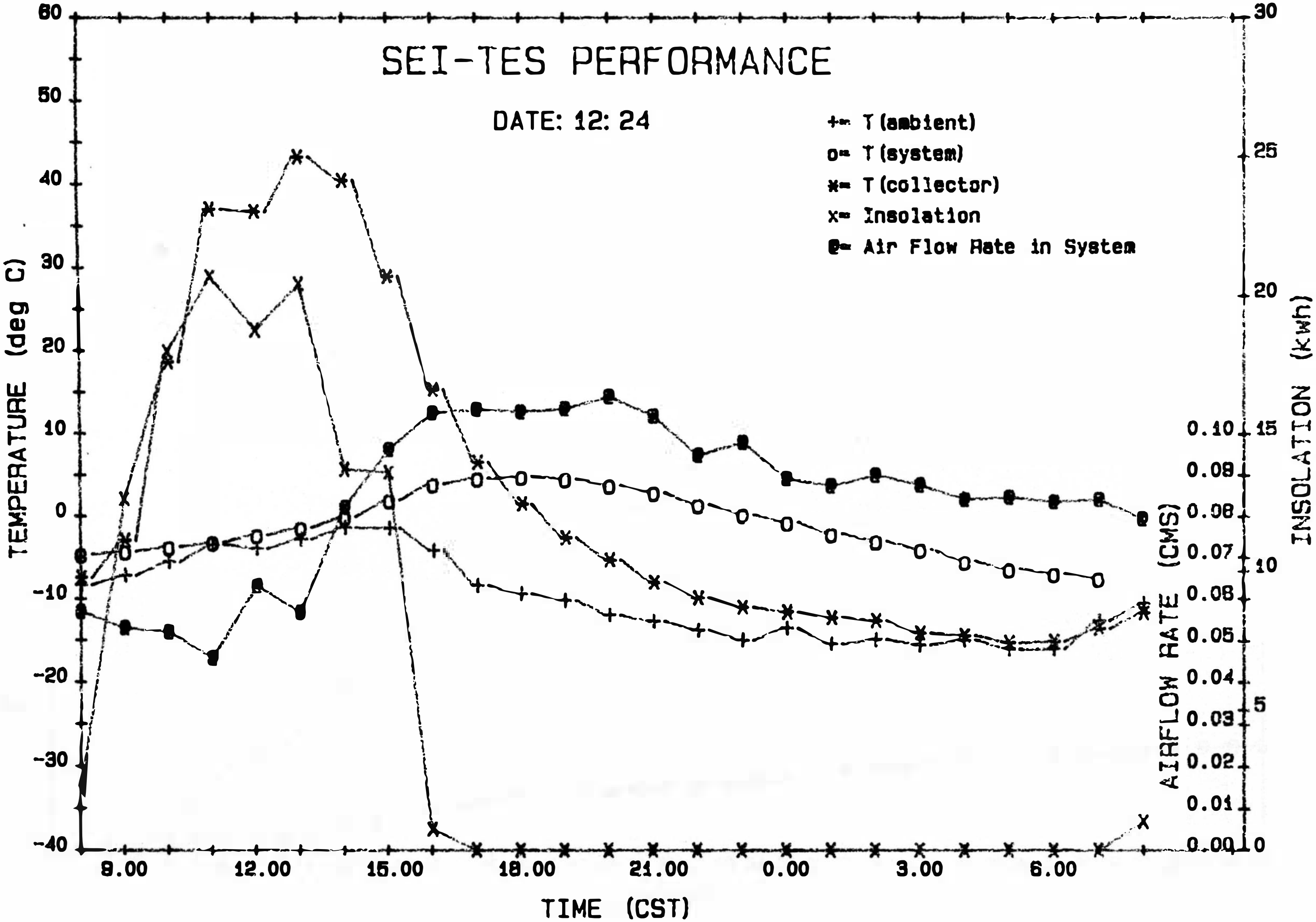


FIGURE 13. HOURLY SEI-TES SYSTEM PERFORMANCE, 24 DECEMBER, 1981.

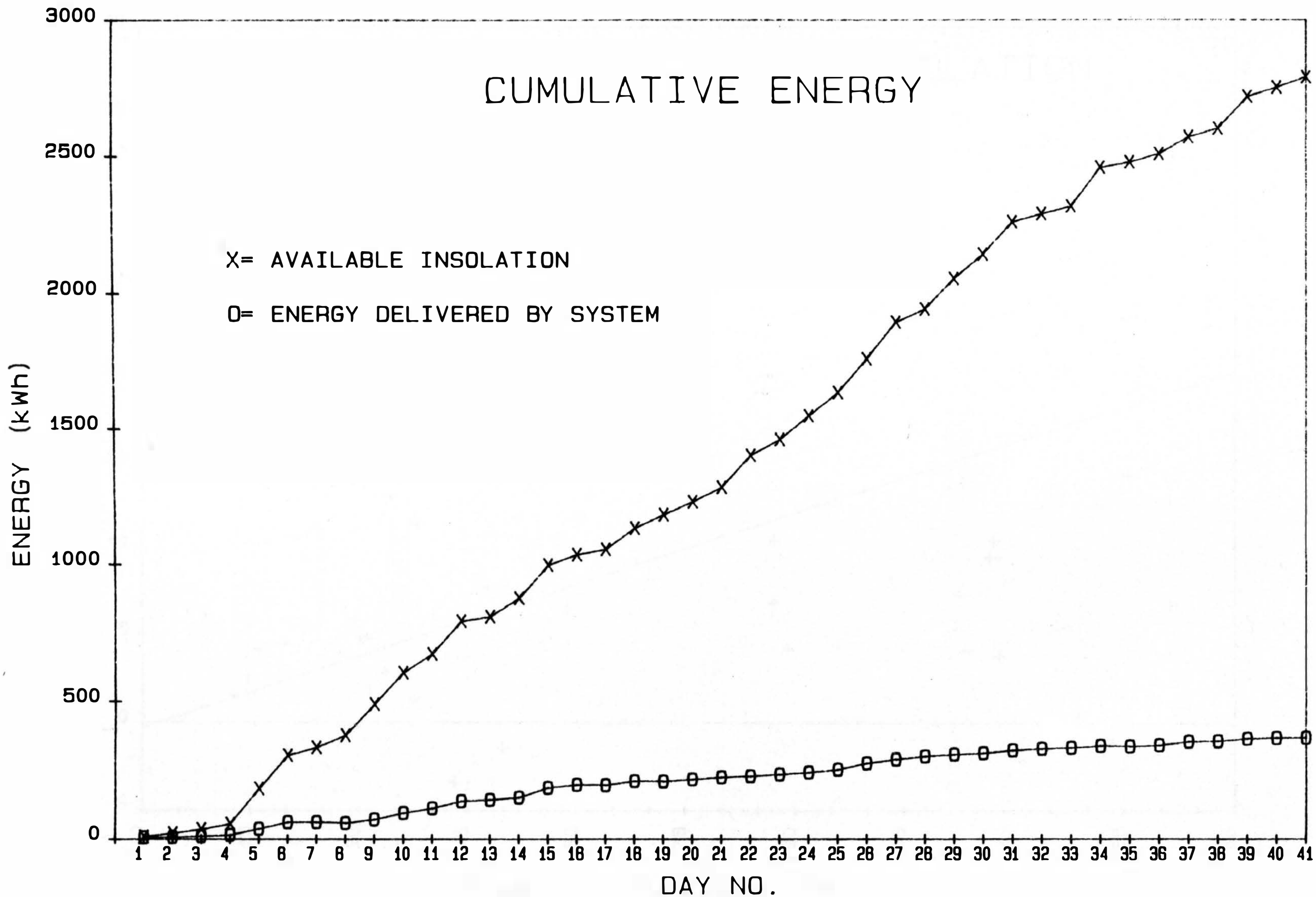


FIGURE 14. PLOT OF CUMULATIVE ENERGY AVAILABLE AND ENERGY DELIVERED BY THE SEI-TES.

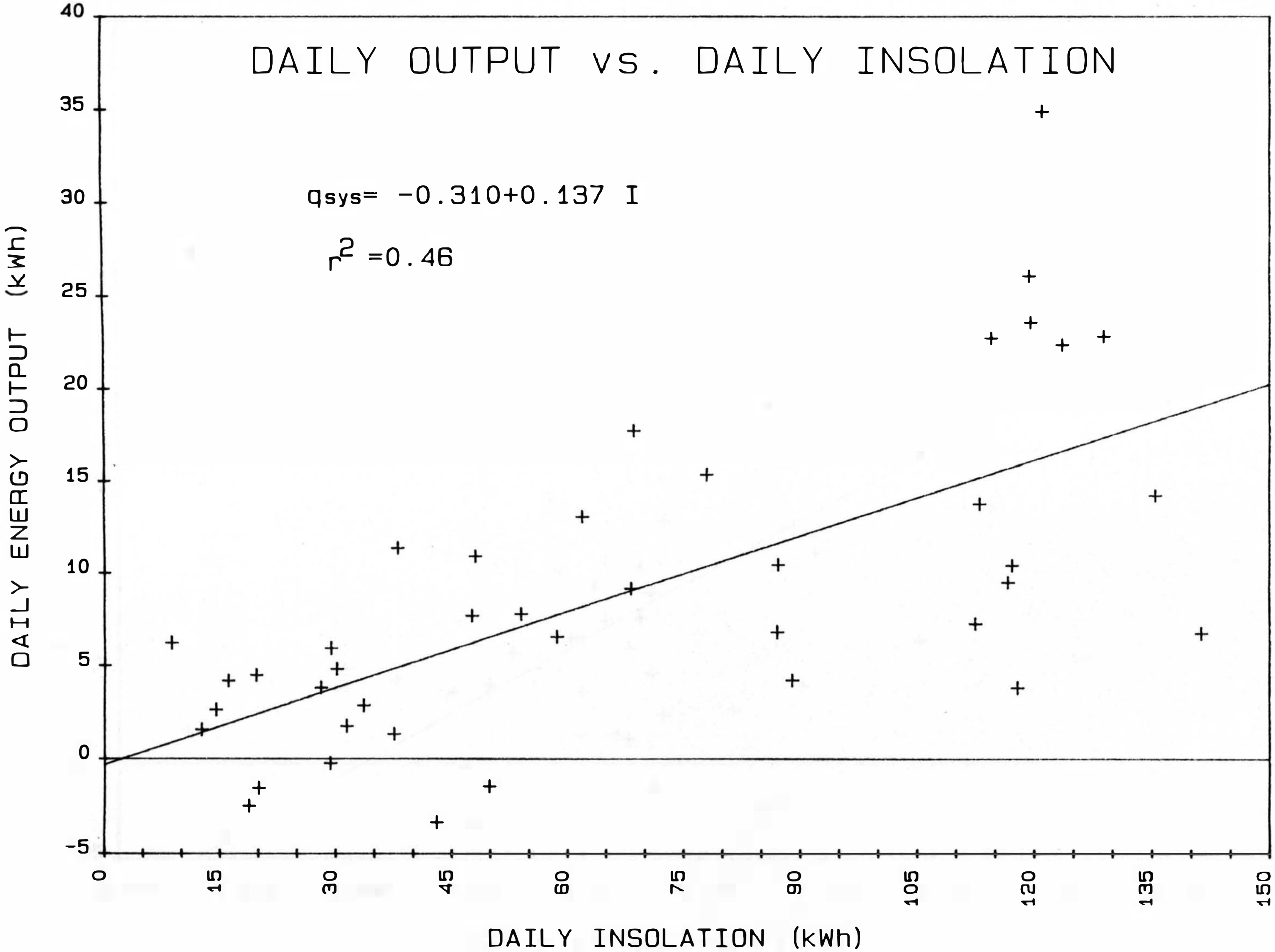


FIGURE 15. DAILY SYSTEM ENERGY OUTPUT vs. AVAILABLE SOLAR RADIATION.

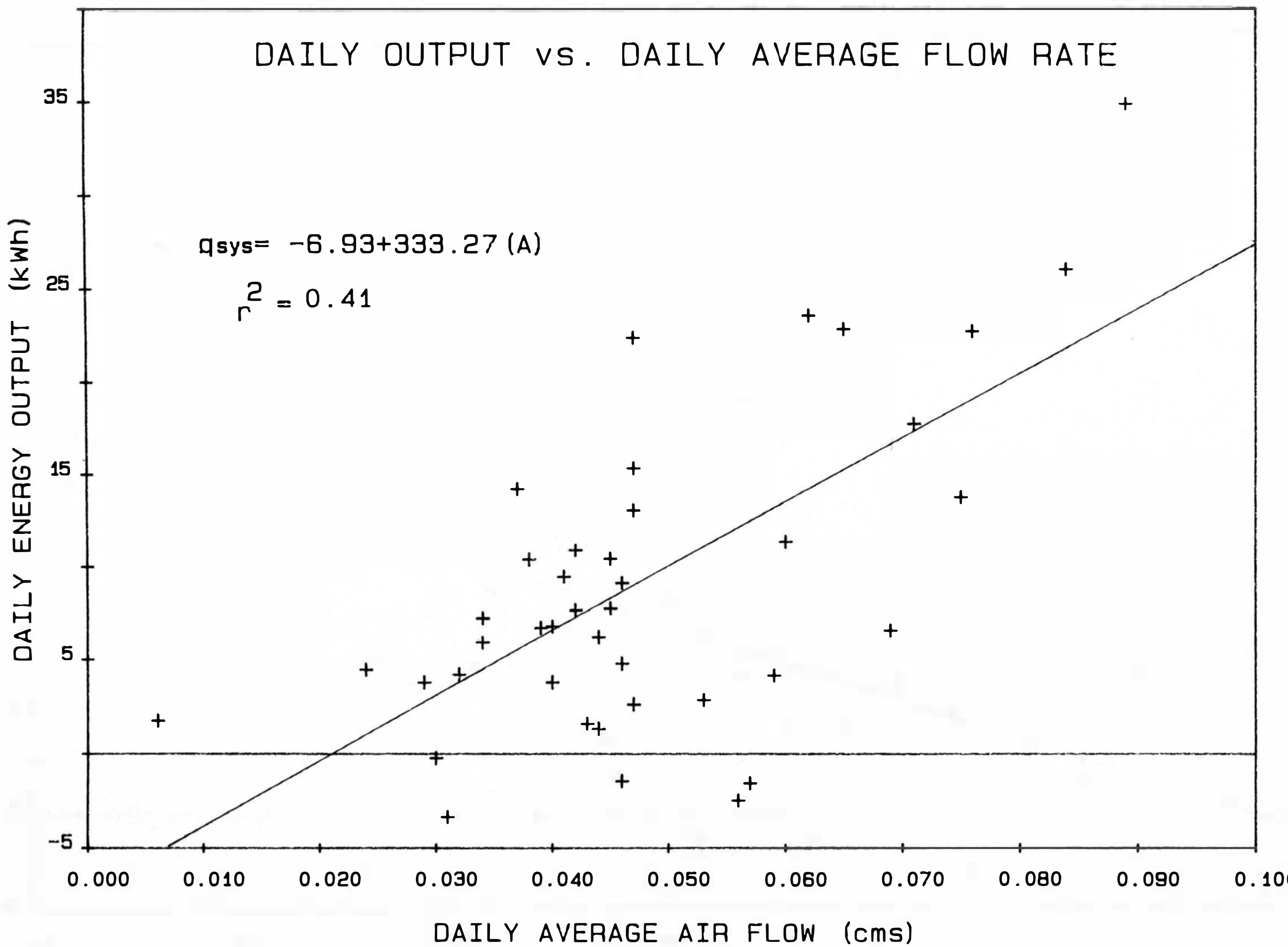


FIGURE 16. DAILY OUTPUT OF THE SYSTEM vs. AVERAGE DAILY AIR FLOW RATE.

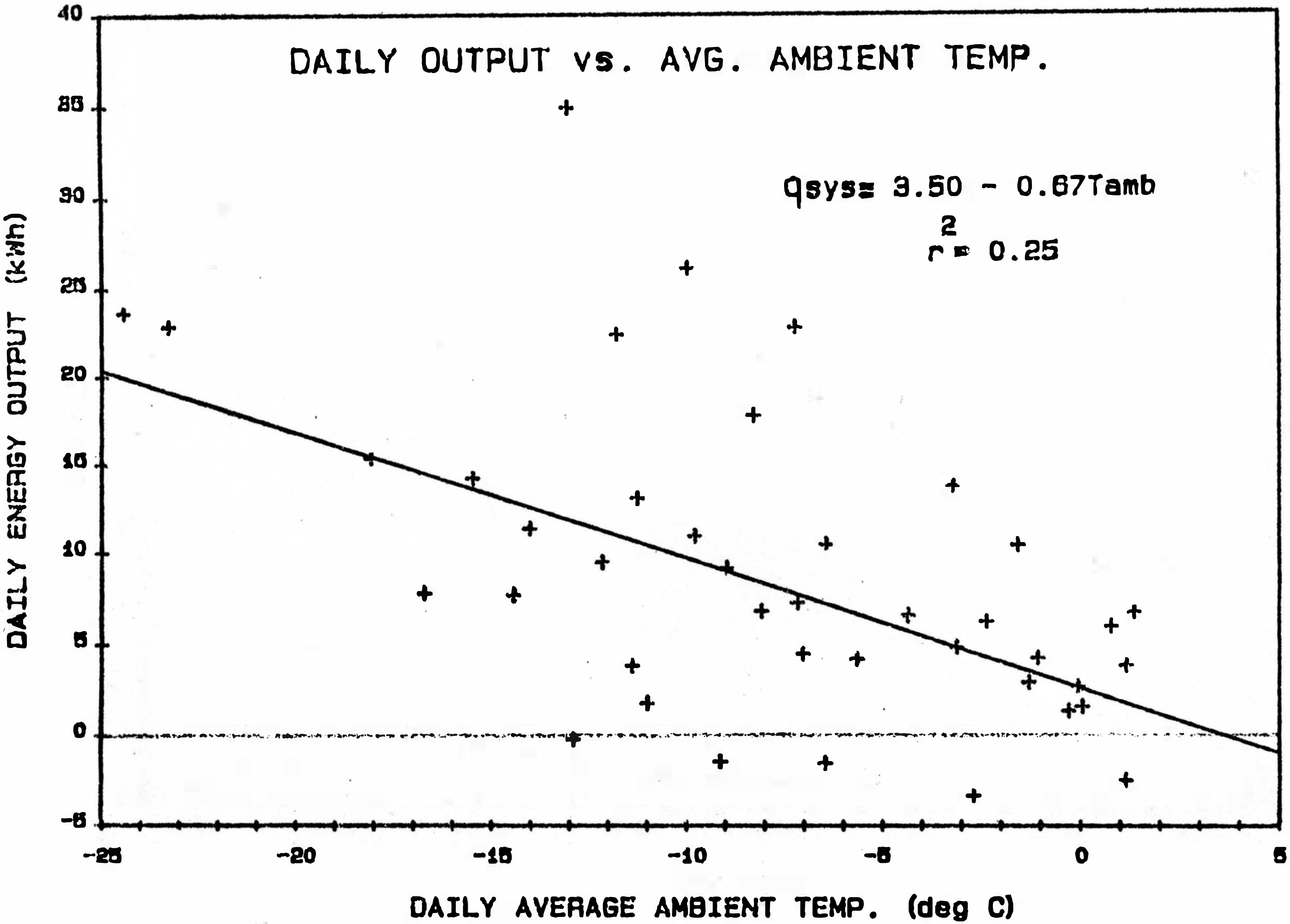


FIGURE 17. DAILY OUTPUT OF THE SYSTEM vs. AVERAGE AMBIENT TEMPERATURE.

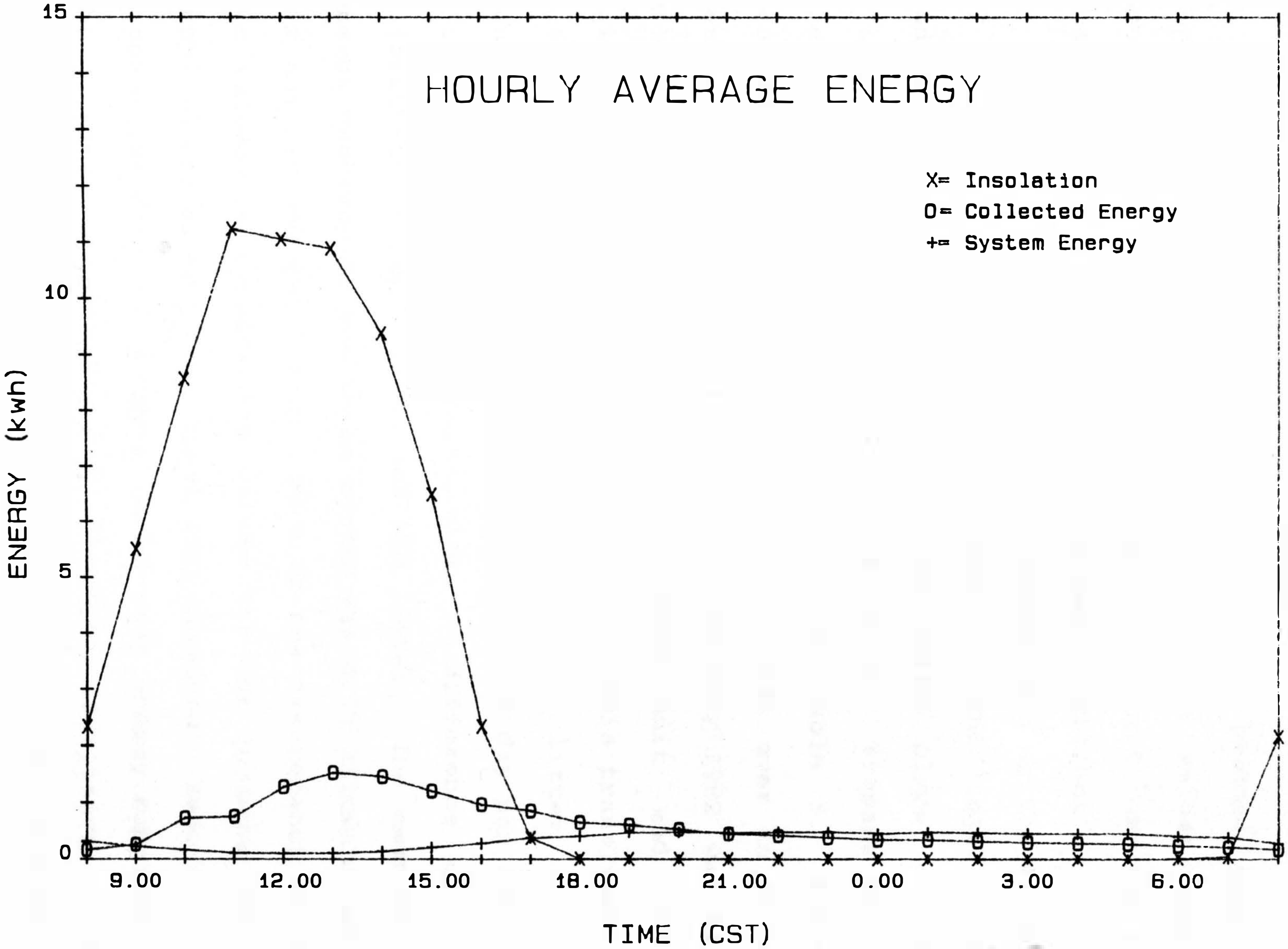


FIGURE 18. AVERAGE ENERGY AVAILABLE, COLLECTED, AND DELIVERED FOR EACH HOUR.

Energy Use Comparisons

Insolation levels during the 1981-83 heating seasons averaged 78 percent of normally expected values, while outside temperatures were near normal. Table 4 compares the observed weather conditions with normal conditions.

Total energy provided by the system during the 41 days of testing was 369.9 kilowatt-hours. The total solar radiation available on the south-facing plane of the collector surface was 2790 kilowatt-hours. Propane energy use in the two units is recorded in Table 5. Energy provided by the conventional heating system over an 81-day period from 4 December 1981 to 23 February 1982 was 5498 kilowatt-hours in the solar-assisted unit and 10771 kilowatt-hours in the conventional unit. This translated to 7.4 kilowatt-hours per day per sow and litter in the conventional unit and 5.7 kilowatt-hours per day per sow and litter in the solar unit, with a difference of 1.7 kilowatt-hours per day per sow and litter. The mean daily energy provided by the solar system was 0.75 kilowatt-hours per day per sow and litter. Part of the discrepancy in the two values is because the values for the propane energy supplied are based on the total fuel consumed. Reducing the propane use data to estimate useful heat energy supplied by the propane and assuming the conversion efficiency of the combustion process to be approximately 0.60, the difference

Table 4 Weather Summary.

	1981-82 Data Collection Period						1983 Data Collection Period	
	December		January		February		February	
	Measured	Normal	Measured	Normal	Measured	Normal	Measured	Normal
Average Ambient Temp., °C	-8.5	-6.1	-11.6	-9.3	-6.7	-7.2	-6.2	-7.2
Average Maximum Ambient Temp., °C	-4.7	-1.0	-7.0	-3.8	-4.3	-1.6	-1.1	-1.6
Average Minimum Ambient Temp., °C	-9.9	-11.2	-15.7	-14.9	-13.9	-12.8	-12.4	-12.8
Daily Insolation, kWh *	63.0	102.9	73.25	97.78	87.8	76.6	54.8	76.6

* Normal daily insolation is based on 50% probability of receiving at least the given amount (Baker and Klink, 1975). Values were converted to a 60° surface with direct normal percentage from List (1966) and tilt factors from Kreith and Kreider, (1978).

Table 5 Propane energy use.

	Total	Solar Unit		Conventional Unit	
	cubic feet, propane gas	cubic feet, propane gas	kWh per * sow & litter	cubic feet, propane gas	kWh per * sow & litter
12/4/81-12/10/81	900	100	6.27	800	33.48
12/10/81-1/5/82	9500	3400	213.41	6100	155.26
1/5/82-1/26/82	7800	2290	143.74	5510	230.57
1/26/82-2/8/82	2000	1110	69.68	890	37.24
2/8/82-2/9/82	300	100	6.27	200	8.37
2/9/82-2/16/82	700	300	18.75	400	16.74
2/16/82-2/23/82	400	0	0	400	16.74
1/29/83-2/15/83	33	10	62.80	23	96.24

* Based on 0.753 kWh/ft^3 of propane gas.

in propane use between the solar-assisted and conventional units would be 1.0 kilowatt-hour per day per sow and litter.

Based on the savings in fuel recorded, all energy supplied by the system during the data collection period was utilized as productive heat in the farrowing building. It should be noted that data were collected during the coldest part of the heating seasons and that the system output was below normally expected levels.

Economic Analysis

The cost of the SEI-TES system, excluding ductwork, was $67.70 per square meter (based on effective area). The cost of extra fans and the amount of ductwork necessary would be variable, depending on individual applications.

Hellickson, et. al. (1981b) found that the utilization factor for the system was about 0.67. The utilization factor is the percentage of the energy supplied by the system that is utilized as productive heat. They based their estimate on the ratio of propane energy saved to energy provided by the solar system. From that study, it was concluded that the utilization factor would be higher under normal weather conditions. Based on the same method of estimation, all of the energy output appeared to have been useful during the 1981-83 investigation. It was concluded that the actual utilization factor was near 1.0. Assuming a system efficiency of 14 percent, a utilization factor of 0.95, and 85 kilowatt-hours per day of available insolation for a 150 day heating season, the system could be expected to save fuel equivalent to about $45.00 per season based on propane at $0.19 per liter (7.1 kilowatt-hours of available energy per liter of liquid propane), or $85.00 per season based on electricity at $0.05 per kilowatt-hour. This is a savings of $2.35 per square meter of effective area per season for a propane heating system or $4.45 per square meter for an electric heating system. Had the system operated at an efficiency of 25 percent, with the utilization factor of 0.67 found by Hellickson, et. al. (1981b), the savings would have been $3.00 per square meter for a propane heating system or $5.60 per square meter for an electric heating system.

System Reliability

Collector

The effects of low air flow rates were discussed in previous sections. Air bypassed the system and entered the building through other channels despite the care taken during construction to insure that all ductwork was airtight. The system performed as expected during the first year of operation, but weathering and shrinking of duct and seal materials allowed infiltration to reduce the effectiveness of the system during subsequent heating seasons.

The design of the collector required an airtight seal between the glazing and absorber at the base of the collector. This prevented drainage of moisture from melted frost and snow between the glazing and absorber. The result was serious rust and corrosion problems in the lower frame and on the lower half of the absorber plates. This corrosion could be expected to reduce the effective life of the system as well as inhibit the ability of the collector to absorb available solar radiation.

Dust and dirt on the collector glazing were not a serious problem. Snow and frost accumulations on the collector surface accounted for some reduction in collector efficiency. Winter storms with even minor amounts of precipitation, accompanied by northwesterly winds, sometimes covered all or portions of the collector/storage unit with snow.

Reflector Materials

One week after the termination of data collection for the 1981-82 season the YS-91 reflective material was destroyed by high winds. It had been in use for one year. It was replaced for the 1982-83 heating season with polished aluminum sheets. These sheets had previously been used on reflectors at Brookings, South Dakota, testing the performance of the SEI-TES system as a water heater. During the summer, the reflective material was damaged by hail which left the entire surface marked with small round dents.

It was hoped that data could be gathered to compare the performance of the aluminum with that of the YS-91. The damaged material continued to reflect a uniform band of radiation onto the north surface of the collector; however, the intensity of the band may have been diminished by the surface irregularities in the aluminum.

The average collector efficiency for the 1982-83 season was 10.9 percent and the average insolation available was 47.1 kilowatt-hours per day. Collector efficiency averaged 19.1 percent for the previous season, with an average of 72.3 kilowatt-hours of radiation available per day. There was also a slight difference in average air flow rate. During the 1981-82 season, the flow rate averaged 0.049 cubic meters per second. For the following heating season, the flow rate averaged 0.042 cubic meters per second. The result was a one percent reduction in the average heat transfer coefficient from the 1981-82 season.

There was a significant difference in collector efficiencies. However, the large difference in average available insolation made it difficult to determine the effect of the damaged reflector. Since the relationship between insolation and collector efficiency was not well established at the measured flow rates, no reliable conclusions could be drawn regarding reflector performance.

DESIGN RECOMMENDATIONS

The three major problems which affected the performance and/or the useful life of the solar heating system were: low air flow rates, snow and moisture accumulation around the collector, and corrosion of metal parts in the collector.

The most practical solution to the air flow problem would be the addition of a fan in the duct between the solar system and the application. This fan would provide the minimum flow for efficient solar system performance. The fan would be needed for livestock buildings with negative pressure ventilation systems. A fan would increase the initial cost of the system, but the data indicate that without it, the reduction in efficiency could make the preheating of ventilation air impractical. Connecting the solar ductwork directly to the building fan for positive pressure ventilation systems would ensure proper fluid flow rate without the need for a supplementary fan.

The inclusion of a fan as part of the system design might be beneficial in other ways. Solar storage air could be ventilated to the outside environment and outside air could be pulled into the building for ventilation whenever outside air temperatures exceeded solar storage temperatures. This could be accomplished by installing a hinged baffle in the duct, controlled manually or with a differential thermostat. This would improve the system efficiency while constant operation of the fan would allow the storage to continue to be charged with the warmer outside air.

The use of a separate fan would also allow flexibility in the storage capacity of the rock bed. Packing density could be increased with smaller diameter rocks, thus increasing the total thermal capacity without altering the basic design of the system. The fan would then be sized for the appropriate flow rate and pressure drop. Flexibility was limited with the present design because of the need to avoid excessive pressure drop across the building ventilation fans.

It should be noted that increased thermal storage capacity would provide a longer time interval between maximum collector temperature and maximum system output temperature. Increased thermal capacity would also tend to level the output temperature.

System placement and surrounding landscape were responsible for the snow accumulations around the collector. The solar heating system was located on the north and west of the swine facility with an open field beyond the system. A windbreak or snow fence on the northwest side would reduce the problem of drifting snow.

System placement is site-specific according to the existing layout of the application. Snow accumulation patterns should be considered when determining the merits of installing an SEI-TES system.

It would be difficult to provide suitable drainage for moisture accumulations in the collector without altering the present air flow pattern. More recent solar units were constructed using galvanized steel on the collector frame, which should minimize the rusting problem.

CONCLUSIONS

The following were concluded regarding the performance of the solar energy intensifier-thermal energy storage (SEI-TES) system in livestock ventilation preheating applications:

1. SEI-TES performance was adversely affected by low air flow rates. A supplemental fan is recommended for negative pressure ventilation applications.
2. Efficiency of the SEI-TES system for ventilation air preheating was 13.8 percent. Previous studies found system efficiency between 24 and 27 percent.
3. Average solar system temperature rise was 5.2 degrees Celsius above ambient.
4. The time lag between maximum energy collected and maximum system output averaged 7 hours.
5. Metal parts in the collector assembly were subject to corrosion due to condensation between the glazing and absorber. Use of galvanized components and rust-resistant coatings may improve system durability.
6. Based on system efficiency of 14 percent, 85 kilowatt-hours per day available insolation for a 150 day heating season, and a 0.95 utilization factor, the system could be expected to save $45.00 per

season with propane fuel at \$0.19 per liter or \$85.00 per season with electricity at \$0.05 per kilowatt-hour. Cost of the 19.1 square meter system, excluding ductwork, was \$67.70 per square meter of effective area.

7. A statistically significant relationship for predicting system output based on air flow rate, ambient temperature, and insolation was developed:

$$TOTAL = -11.9 + 250.0 AIRF - 0.395 T_{amb} + 0.086 INSOL$$

SUMMARY

Solar heat is well suited to many agricultural applications because of the efficient match of high entropy levels in both the source and the systems receiving the energy. Three processes have been identified as the most promising situations requiring low temperature heat: in-storage grain drying, ventilation air preheating for livestock housing, and service hot water heating. Because a solar system which is versatile in its applicability and can remain in use throughout most of the year has the most favorable economic potential, a multi-use solar system was designed at South Dakota State University and tested in a number of applications and locations.

The system was installed at a swine farrowing facility near Sioux Falls, South Dakota and field performance tests were conducted during the 1981-82 and 1982-83 winter heating seasons. Low air-flow rates due to infiltration were encountered, and the system performed below expected levels. The system was instrumented for data collection and daily performance of the system was analyzed statistically. A relationship was developed to predict the daily system output.

Overall system efficiency averaged 13.8 percent during the test periods, which is below the 24 to 27 percent

encountered in previous tests. A fan in the system duct was recommended to improve system performance in applications where infiltration might alter the design flow rates. Corrosion resistant coatings for certain components were recommended in order to extend the life of the solar system to an economically acceptable level.

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Appendix A

FARROWING COMPLEX PLANS

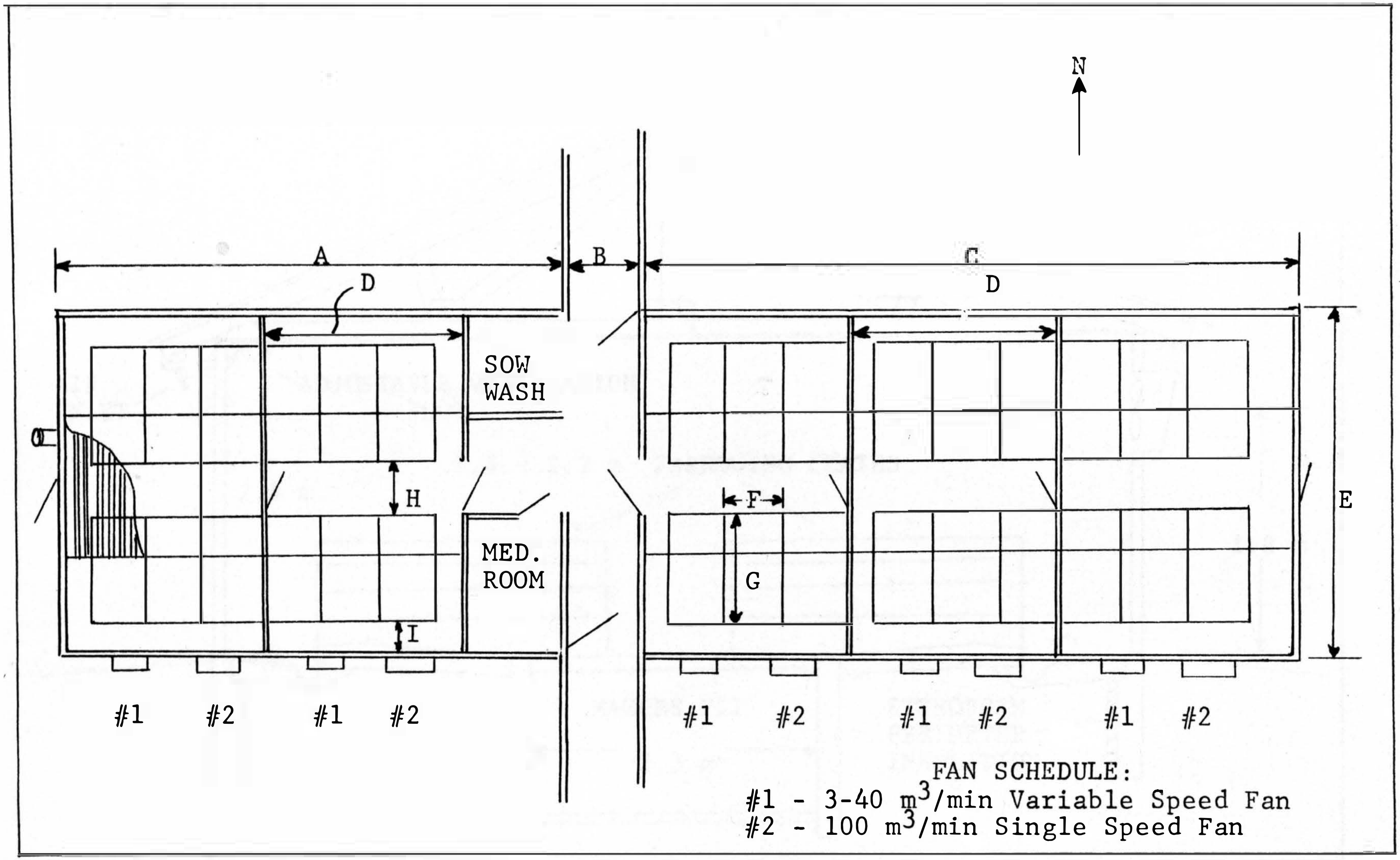

FIGURE A1. PLAN VIEW OF THE FARROWING FACILITY.
DIMENSIONS: A= 12.2 m, B= 1.8 m, C= 14.9 m, D= 4.9 m, E= 7.3 m, F= 1.4 m, G= 2.3 m, H= 0.9 m, I= 0.8 m.

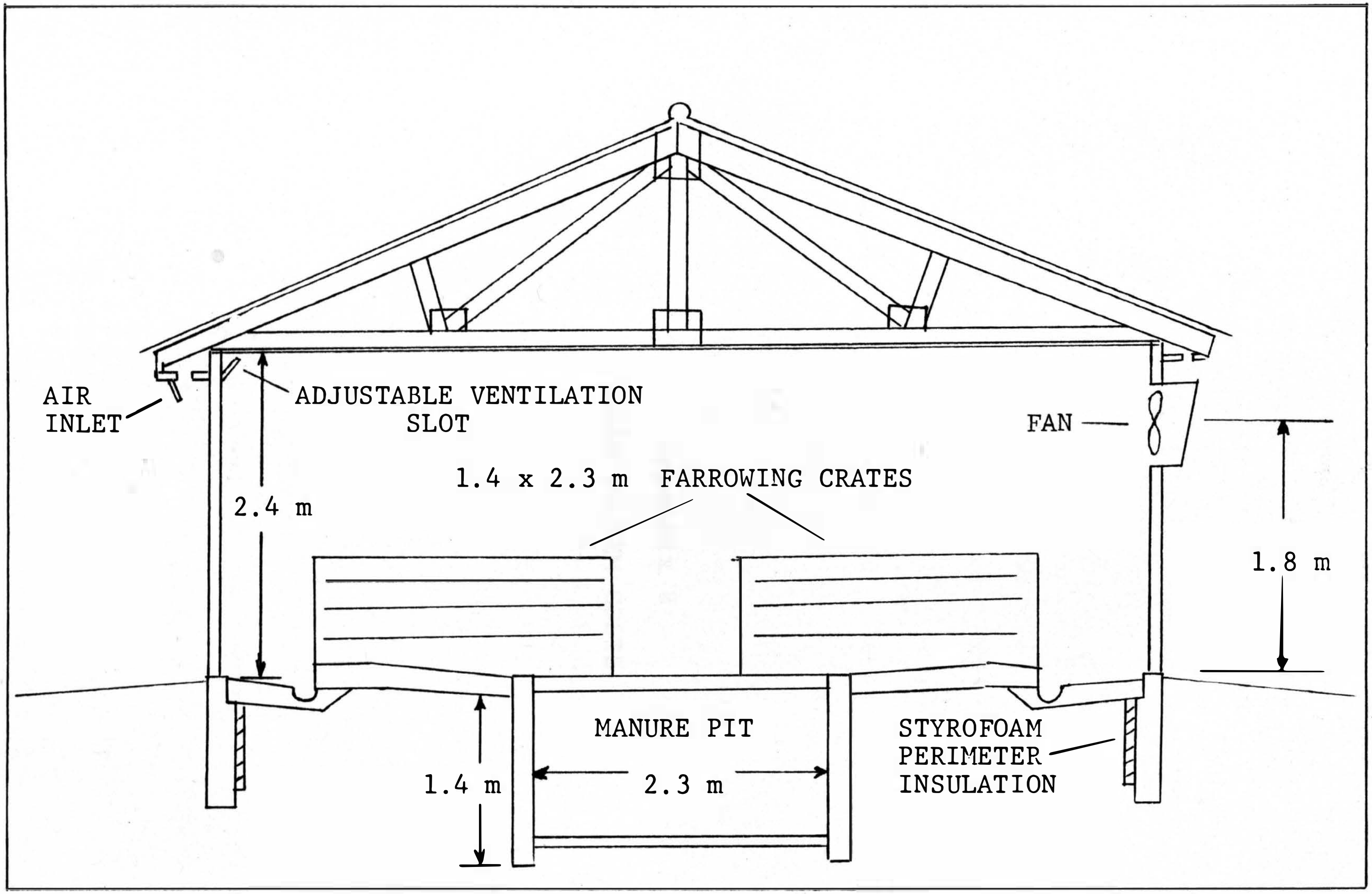

FIGURE A2. SIDE VIEW OF FARROWING FACILITIES.

Appendix B

SOLAR SYSTEM COSTS

Table B.1 Cost of materials for SEI-TES system. *

Quantity	Description	Cost
	Collector	
17.65 m^2	1.2 mm sheet steel (absorber plate)	
4.46 m^2	1.2 mm sheet steel (cover)	
2.97 m^2	1.2 mm sheet steel (top)	
0.93 m^2	1.2 mm sheet steel (ends)	$185.00
40.80 m	1.9 x 1.9 cm sq. tubing	45.00
2.10 m	0.6 x 1.9 cm strap	2.00
0.18 m^2	1.2 mm tin (glass supports)	1.00
15.24 m^2	1.3 cm plywood (backing)	62.00
1	1.22 x 2.44 m sheet of 2.5 cm styrofoam (top)	5.00
0.74 m^2	5.0 cm styrofoam (ends)	3.00
17.65 m^2	low iron glass panels	285.00
10	1.2 mm, 3 m long, 9 cm wide bent w/2.3 cm id.	26.00
10	1.2 mm, 3 m long, 9 cm wide bent w/3.5 cm id.	23.00
20	1.2 mm, 3 m long, 2.5 cm wide bent to 1.3 x 1.3 cm L	10.00
	Labor - shearing and bending	90.00
	Reflector	
25.9 m	0.5 x 1.9 cm strap	18.00
109.7 m	1.3 cm sq. rod	135.00

Table B.1 (cont.)

Quantity	Description	Cost
116.7 m	1.3 cm rod	$113.00
13.7 m	2.5 cm rod	53.00
1.5 m	1.9 x 0.5 cm angle	2.00
1.5 m	1.0 x 0.5 cm strap	1.00
51.8 m	1.9 x 0.3 cm strap	24.00
39.5 m^2	Kinglux reflector material	850.00
10.0 m	10 cm flange I-beam	112.00
14.6 m	5 x 10 cm reflector support	12.00
18.3 m	15 cm beam	90.00
	Miscellaneous Hardware	
80	7.6 x 0.6 cm bolts (reflector). . .	4.00
40	10 x 0.6 cm threaded rod (collector). . .	2.60
80	0.6 cm washers	2.10
200	0.6 cm nuts	6.00
80	0.5 cm nuts	1.60
80	0.5 x 1.3 cm machine screws.	1.60
16	suitcase clasps	6.40
30	0.6 x 2.5 cm bolts	1.20
18.3 m	1.9 x 0.3 cm foam (single stick). . . .	3.00
21.9 m	1.3 x 1.3 cm foam (single stick)	5.70
51.8 m	0.08 x 1.9 cm foam (double stick). . . .	8.20

Table B.1 (cont.)

Quantity	Description	Cost
125	#8 x 1.9 cm wood screws	$3.75
24	17.3 x 1.3 cm bolts & nuts	13.00
	Drying Base	
4	1.3 cm plywood sheets	48.00
39 m	5 x 10 cm lumber	32.00
4	5 cm styrofoam sheets	48.00
1.5 m^2	light tin (wood shield)	7.00
	Rock Storage Base	
21	1.3 cm plywood sheets	252.00
4	5 x 15 cm boards, 4.9 m long	19.00
30.2 m^2	15 cm fiberglass insulation	90.00
	Water Heating Materials	
	Fan, heat exchanger, plumbing, etc. . .	650.00
	Total Cost (for all three applications).	3352.00

* Based on 1982 prices. Cost of machining and partial assembly are included.

Appendix C

MULTIPLE REGRESSION TABLE

Table C.1 Multiple regression table for TOTAL versus AIRF, T_{amb}, and INSOL.

	DF	SUM OF SQUARES	MEAN SQUARE	F	PROB F
Regression	3	2240.75	746.92	39.41	0.0001
Error	37	701.30	18.95		
Total	40	2942.05			
	B Value*	Std. Error	Type II SS	F	Prob F
Intercept	-11.87				
AIRF	250.00	43.20	634.73	33.49	0.0001
T_{amb}	-0.39	0.11	223.49	11.79	0.0015
INSOL	0.09	0.02	417.00	22.00	0.0001
	r squared = 0.76				

* The B Value is the coefficient of the variable listed.

Appendix D

DAILY DATA SUMMARIES

Table D.1 Daily data summaries.

Date	INSOL (kWh)	T_{amb} (°C)	AIRF (m^3/s)	COLL (%)	SYS (%)	TOTAL (kWh)
1981						
12:10	8.68	-2.4	0.044	51.6	71.6	6.21
12:11	12.61	0.1	0.043	12.1	12.6	1.58
12:12	14.48	-0.1	0.047	14.6	18.2	2.63
12:13	19.69	-7.0	0.024	19.0	22.7	4.47
12:17	128.88	-23.3	0.065	24.5	17.7	22.85
12:18	119.57	-24.4	0.062	29.5	19.8	23.61
12:19	29.32	-12.9	0.030	5.4	-0.8	-0.22
12:20	43.08	-2.7	0.031	0.2	-7.8	-3.36
12:21	113.02	-3.2	0.075	21.7	12.2	13.76
12:22	114.53	-7.2	0.076	25.3	19.9	22.75
12:23	68.56	-8.3	0.071	26.7	25.9	17.74
12:24	119.47	-10.0	0.084	24.0	21.8	26.08
12:26	16.07	-5.7	0.059	13.3	25.9	4.17
12:27	68.17	-9.0	0.046	20.6	13.4	9.15
12:28	121.10	-13.0	0.089	34.9	28.8	34.90
12:29	37.99	-14.0	0.060	14.3	29.9	11.37
12:30	20.00	-6.5	0.057	5.0	-7.6	-1.53
12:31	77.95	-18.1	0.047	10.3	19.7	15.35
1982						
01:01	49.84	-9.1	0.046	6.7	-2.9	-1.44
01:02	47.65	-14.4	0.042	13.2	16.1	7.68
01:03	53.95	-16.7	0.045	16.3	14.4	7.77
01:04	117.89	-11.4	0.029	14.2	3.2	3.80

Table D.1 (cont.)

Date	INSOL (kWh)	T_{amb} (°C)	AIRF (m^3/s)	COLL (%)	SYS (%)	TOTAL (kWh)
01:27	58.58	-4.4	0.069	23.0	11.2	6.55
01:28	87.01	-8.1	0.040	19.4	7.8	6.80
01:29	87.09	-6.4	0.045	27.5	12.0	10.45
02:03	123.60	-11.8	0.047	24.5	18.1	22.39
02:11	135.44	-15.5	0.037	22.1	10.5	14.22
02:12	48.02	-9.8	0.042	23.9	22.7	10.92
02:13	112.47	-7.2	0.034	19.4	6.4	7.24
02:14	88.97	-1.1	0.032	15.8	4.7	4.22
02:15	117.15	-1.6	0.038	19.8	8.9	10.41
02:16	29.38	0.8	0.034	18.4	20.2	5.93
02:17	28.08	1.2	0.040	10.3	13.5	3.80
02:18	141.26	1.4	0.039	22.0	4.8	6.72
1983						
01:28	18.73	1.1	0.056	11.6	-13.2	-2.48
01:29	30.13	-3.1	0.046	7.2	16.0	4.81
01:30	61.85	-11.3	0.047	22.1	21.1	13.06
02:02	31.42	-11.0	0.006	5.4	5.6	1.77
02:03	116.66	-12.2	0.041	7.4	8.1	9.48
02:11	33.64	-1.3	0.053	13.6	8.5	2.87
02:12	37.56	-0.3	0.044	9.0	3.6	1.34