

UTILIZATION OF A WASTE TREATMENT  
LAGOON AS AN ENERGY SOURCE

by

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## INTRODUCTION

The importance of fossil fuels on today's modern farm has been well documented. Energy is used in the production of both crops and livestock. Energy is needed to power machinery and produce input materials such as pesticides, fertilizers and seeds for crop production. In livestock production, energy is used to prepare and transport feed and to provide lighting and ventilation, as well as for many other uses. If heated areas are provided, space heating is usually the major energy cost.

Swine production in the United States consumes  $2.4 \times 10^{15}$  J ( $2.3 \times 10^{12}$  BTUs) yearly (Hughes, Hill and Vaughan, 1976). Most of the energy is used in the farrowing and nursery areas where young pigs are housed. Although older pigs can cope with colder temperatures, new born pigs are very vulnerable to temperatures below  $29.4\text{--}32.2^{\circ}$  C ( $85\text{--}90^{\circ}$  F).

The temperature in the farrowing house is kept slightly below  $21.1^{\circ}$  C ( $70^{\circ}$  F) to suit the sow so localized supplemental heat is required for new born pigs. Supplemental heat can be provided by heat lamps or catalytic brooders. The supplemental heaters are placed so that the pigs can move around under them to locate their individual comfort zones.

In the nursery, the pigs are gradually subjected to

lower temperature levels as they get older. After adjustment to lower temperatures, the pigs may be moved to outside lots or unheated confinement buildings for finishing.

The pigs' susceptibility to cold temperature calls for a continuous supply of energy to the farrowing house and nursery. One difference between a profit or loss to the producer is the dependability of the energy supply. With rising fossil fuel costs and since the supply of conventional energy sources has been shown to be subject to interruption, an expanded interest in solar energy has developed.

#### Solar Energy for Swine Shelters

While solar energy is considered to be "free", the equipment to utilize this energy is not (Hughes and Vaughan, 1975). Solar energy is not uniform throughout the country and there are also daily variations dependent on the date and weather conditions. For example, insolation on a cloudy, rainy day may be a fraction of that on a clear, sunny day. As a result of the variations in solar insolation rates, an energy storage system is required if solar energy is to be used for heating swine housing.

A typical solar system would consist of the following components (Hughes, Hill and Vaughan, 1976):

1. Solar collectors.
2. Energy storage facility.
3. Heating equipment.
4. Auxiliary heating system.
5. Controls.

Commercially or privately built solar flat plate panels are the most common form of solar collectors. Solar insolation is collected as heat which is used to raise the temperature of water or air. Heat energy may be stored in many ways for later use.

Energy may be stored in water, rocks and heat of fusion materials. Small water tanks, swimming pools, underground tanks or possibly waste treatment lagoons are examples of water storage systems. Beds of gravel, large rock and salt are examples of air storage systems. In any storage system, minimization of heat loss is very important. An increase in fossil fuel costs for supplemental heat results from poor storage management.

Several systems have been proposed to use solar heat in swine facilities. The most common are; 1) hot water lines in the nursery and farrowing room floor, 2) forced air circulation over heated water pipes and 3) a solar assisted heat-pump.

On days when solar insolation is minimal or at night, a pure solar energy system will be inoperable and an

auxiliary heating system is required. Auxiliary heat can be provided by direct combustion of fossil fuels or by electricity. The auxiliary system operates until sufficient solar heat is available to carry the heating load.

An adequate control system is important to avoid constant monitoring of the system. Controls can determine when to collect solar energy and which heating system to operate. A good control system automatically switches from one heating system to another at any time of day or night.

Due to time limitations, only one component of the solar energy system was investigated. The utilization of a waste treatment lagoon as an energy source and storage medium was selected for study. The objectives of this study were:

1. To evaluate the amount of low level energy in a waste treatment lagoon. Low level energy is defined as the energy available when the lagoon temperature is below 37.8° C (100° F). At this temperature, the lagoon water cannot be used for direct heating and energy must be extracted by other means.
2. To determine if the low level energy can be removed and used to heat swine housing.
3. To determine the effect, if any, of energy removal

on the thermal stability of the waste treatment lagoon. Thermal stability was determined by a corresponding temperature change in the lagoon temperature when energy was removed. For this study, a temperature change of 2.8° C (5° F) for the entire lagoon was considered as leading to thermal instability.

## LITERATURE REVIEW

Bodies of water have long been considered as possible sources for trapping solar energy. Claude (1930) and Bleau and Nizery (1930) attempted to utilize the oceans as large solar collectors. Although temperature differences of 20° C (68° F) were found, they were not deemed sufficient to operate practical heating systems. To increase the difference, the water was pumped into shallow, black-bottomed pans. An increase of a few degrees was obtained, but was still not sufficient for the operation of a heating system using a heat engine.

In 1948, Bloch suggested that heat could be collected more effectively if convection could be avoided by using density gradients similar to those found in certain Hungarian and Romanian lakes. For example, the temperature of these lakes reach a maximum of 70° C (158° F) in the summer.

Tabor (1961) discussed the problems of maintaining and economically utilizing the solar pond. Later, Tabor (1964), Tabor and Matz (1965) and Weinberger (1964) documented successful solar ponds. The first pond developed a maximum temperature of 60° C (140° F). By experimenting with different salts and gradients, a maximum temperature of 90° C (194° F) was reached.

Usmanov, Umarov and Zakhidov (1969) discussed the theoretical operation and development of solar ponds in the Soviet Union. They predicted that a temperature of nearly 100° C (212° F) could be developed in a properly built and operated solar pond. Eliseev and Usmanov (1971) reported measurement of a temperature of 94° C (201.2° F) in support of this claim. From 1971 through 1973, temperatures as high as 90° C (194° F) in solar ponds have been consistently documented. A new type of solar pond was developed in 1973 by Usmanov, Eliseev and Umarov. By using multiple layers of thin transparent plastic film in conjunction with density gradients, an increase in temperature of 8% (7.2° C) was realized. The addition of the plastic film made the removal of heat much easier by reducing mixing when the water was pumped through a heat exchanger and returned to the appropriate layer.

Dickinson, Clark, Day and Wouters (1976) discussed a system which utilized the idea of a solar pond and collector. The water was pumped through plastic covered shallow ponds and returned to an underground storage tank. The water entered the pond at 70° C (158° F) and exited at 75° C (167° F).

Bridgers, Paxton and Haines (1957) utilized large flat plate collectors to heat a 6000 gallon underground storage tank. The experiment was conducted in Albuquerque,

New Mexico and the water temperature in the tank reached  $21.1^{\circ}\text{C}$  ( $70^{\circ}\text{F}$ ). Heating and cooling of the building was accomplished by a water-to-air heat pump using the water in the tank as a heat sink.

Dehrer (1960) heated a 12,000 gallon swimming pool by the utilization of flat plate collectors. For heating, the water was pumped through the collectors and returned to the uncovered pool. The pool temperature ranged from  $22.8\text{--}23.9^{\circ}\text{C}$  ( $73\text{--}75^{\circ}\text{F}$ ) at 4 A.M. to  $25.6\text{--}26.7^{\circ}\text{C}$  ( $78\text{--}80^{\circ}\text{F}$ ) at 4 P.M. The corresponding air temperatures, recorded during the summer at Fairfax, California, ranged from  $10.0\text{--}14.4^{\circ}\text{C}$  ( $50\text{--}58^{\circ}\text{F}$ ) and  $24.4\text{--}33.3^{\circ}\text{C}$  ( $76\text{--}92^{\circ}\text{F}$ ), respectively.

Czarnecki (1963) utilized an inflated plastic cover to help trap solar insolation for swimming pool heating. Tests were run in September and October of 1960 as well as August through November of 1961. Mean pool temperatures exceeded mean air temperatures for these dates by  $3.8^{\circ}\text{C}$  ( $6.9^{\circ}\text{F}$ ). He presented equations for calculating heat losses due to convection, radiation and evaporation. Using these equations, predictions of heating loads required to keep the pool at a specific temperature were determined.

Hill (1975), February 27 to April 2, 1975, found that temperatures in a manure holding tank varied between  $2.5\text{--}7.2^{\circ}\text{C}$  ( $36.5\text{--}45.0^{\circ}\text{F}$ ), depending on the depth of measure-

ment. The air temperature varied between minus 10 to 24.4° C (14.0-76.0° F) for the same period.

Vaughan, Bell and Hughes (1976) proposed a system utilizing a solar-assisted heat pump with lagoon storage to heat swine housing. The system consisted of a waste treatment lagoon, an energy exchanger, solar collector, a heat pump and a swine nursery. In initial tests, a temperature drop of 2.2-5.6° C (4-10° F) was found when the working medium, 26.7-35° C (80-95° F) water, was pumped through the exchanger. The lagoon temperature at the time was 17.2° C (63° F).

## MATERIALS AND METHODS

To investigate the feasibility of utilizing a waste treatment lagoon as an energy source, a two step study was developed. The first step dealt with the development and testing of an experimental swine heating system utilizing solar energy and the lagoon as heat sources. The second dealt with the development of a computer model to predict available solar insolation and the waste treatment lagoon's response to natural energy losses and artificial energy removal to heat swine housing.

### Experimental Swine Heating System

The project involving the experimental swine heating system was located at the VPI&SU Swine Center. The main components of the system, as described previously by Vaughan, Bell, Hill and Holmes, were (Figure 1):

1. Pig nursery
2. Heat pump
3. Solar collectors
4. Waste treatment lagoon
5. Lagoon heat exchanger

#### PIG NURSERY

The pig nursery, Figure 2, was 2.4 m (8 ft) wide by 4.9 m (16 ft) long with a height of 2.4 m (8 ft). The building was plywood construction with 0.635 cm (1/4 inch)

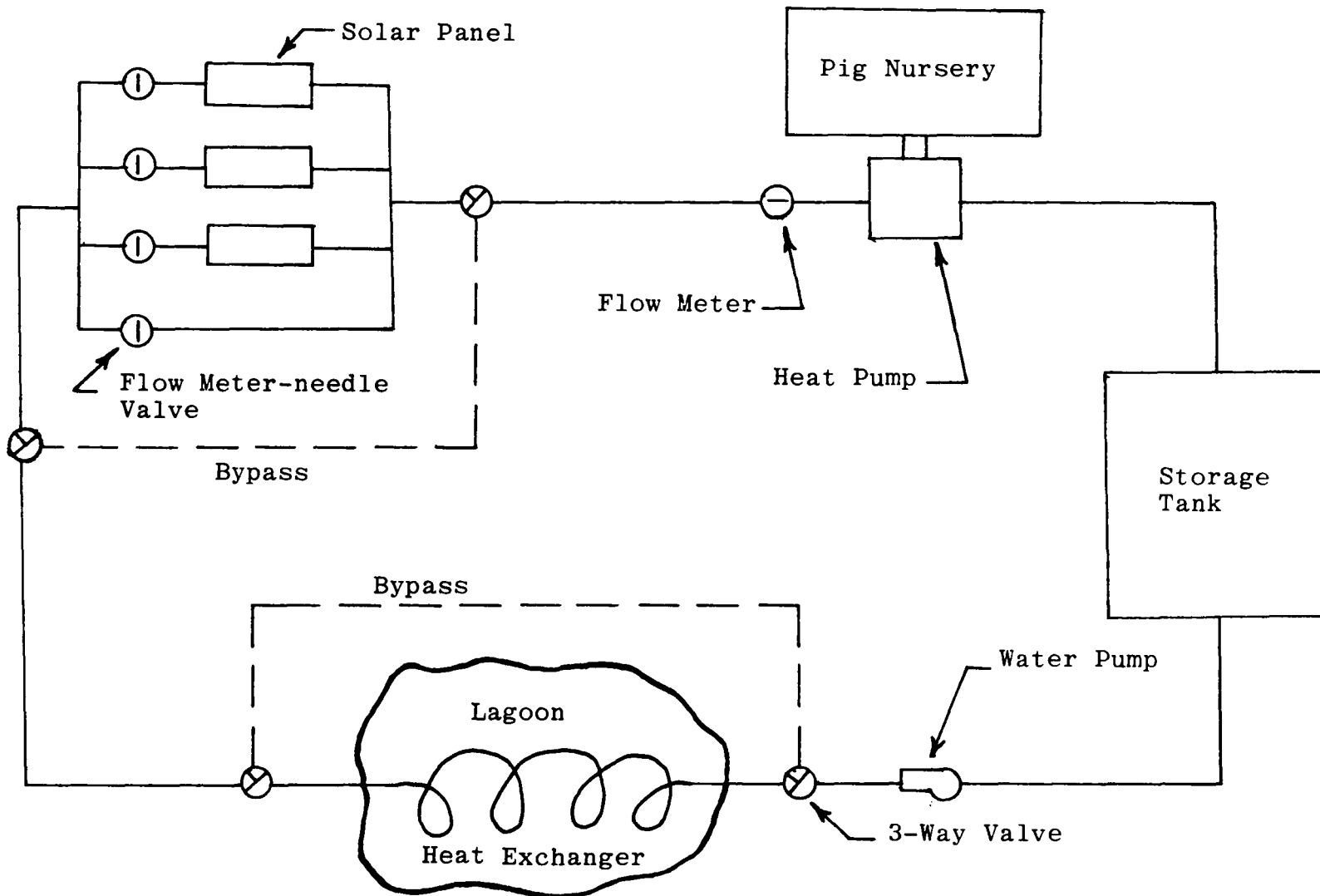


Figure 1. Schematic diagram of the solar assisted heat pump system.



Figure 2. Experimental pig nursery located at the VPI & SU Swine Center.

interior and 1.27 cm (1/2 inch) exterior and floor. Aluminum roofing was used and the building mounted on 10.16 cm x 15.24 cm (4 x 6) pressure treated skids for mobility. The walls, roof and ceiling had 8.89 cm (3.5 inch) of fiberglass insulation.

Ventilation was provided by a thermostatically controlled fan mounted in the wall. During system testing, the fan provided additional loading since pigs were not placed in the building.

#### HEAT PUMP

A Dunham-Bush Model AQM 13 water-to-air heat pump, Figure 3, was selected for heating and cooling the building. The heat pump had a capacity of  $1.4 \times 10^7$  J/hour (13,500 BTU/hour) for heating and  $1.3 \times 10^7$  J/hour (12,000 BTU/hour) for cooling (Dunham-Bush, 1975).

During heating, heat was transferred from the water to the refrigerant as it circulated through the water-to-refrigerant exchanger. At this point, nursery air was then circulated over the coils to transfer the available heat to the building.

During cooling, the refrigerant flow was reversed. Air from the nursery was circulated around the air-to-refrigerant coils and cooled. The heat transferred to the refrigerant was carried to the water-to-refrigerant exchanger where circulating water cooled the refrigerant.

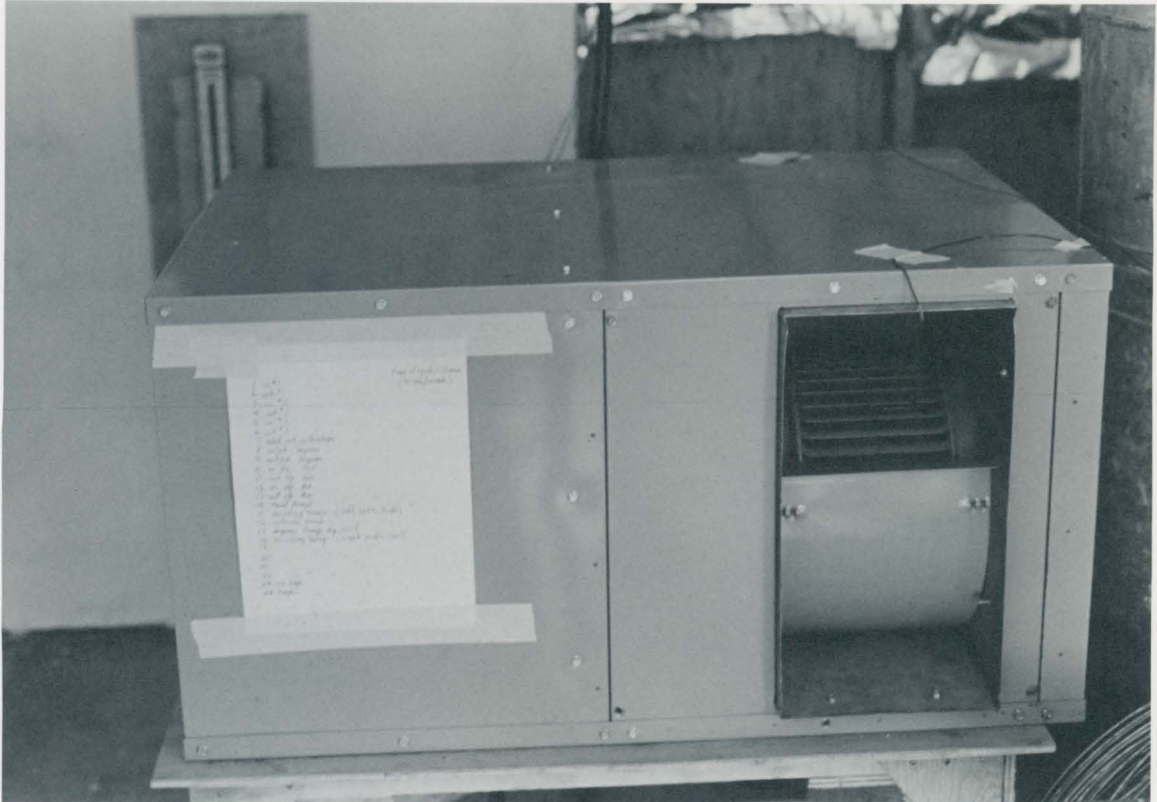


Figure 3. Dunham-Bush Model AQM-13 heat pump used in the experimental solar heating system.

## SOLAR COLLECTORS

Three solar panels were used to supply additional heat to the heat pump. The collectors were the PPG Baseline Solar Collector, the Reynolds Torex Model 14 and the Sunworks Solector Surface-Mounted Module, Figures 4-6, respectively. The PPG model had two glass cover plates separated by an air gap and a roll-bond aluminum collector with a non-selective black absorber. The Reynolds collector had a double-layer of Tedlar film with a small air gap and aluminum tubing in a serpentine configuration. The Tedlar film and aluminum tubing were mounted into a flat-black absorber plate. The Sunworks collector had a single glass cover and a copper absorber with a selective coating (black). The PPG and Sunworks collectors had parallel flow patterns.

The total area of the collector panels was approximately  $7.15 \text{ m}^2$  ( $77 \text{ ft}^2$ ). The collectors were connected in parallel and a flow rate of 2.54 liters/minute (0.67 gallons/min) was used in each collector. To accommodate the requirements of the heat pump, an additional flow of 3.8-7.6 liters/min (1-2 gallons/min) was being pumped through a bypass line. The total combined flow of 11.4-15.1 liters/min (3-4 gallons/min) was sufficient for operation of the heat pump.

To provide the required flow, a variable flow water



Figure 4. Installed PPG collector at the VPI & SU Swine Center.

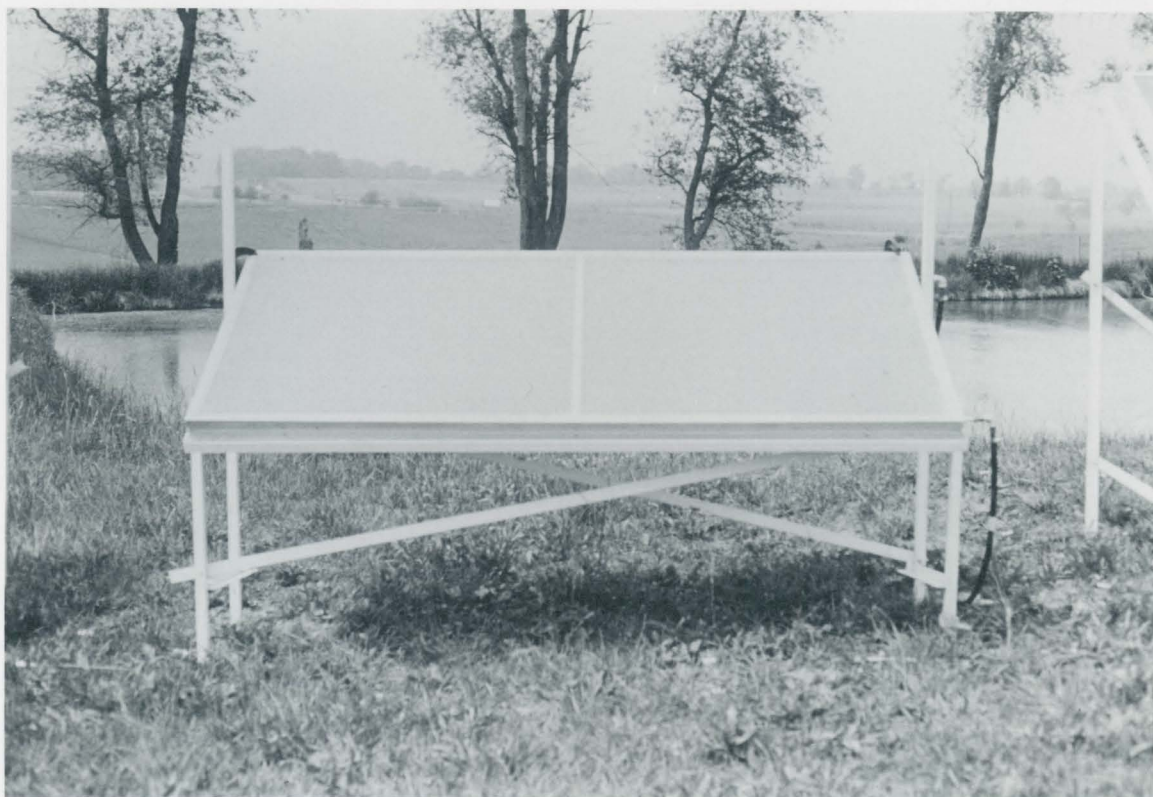


Figure 5. Installed Reynolds collector at the VPI & SU Swine Center.



Figure 6. Installed Sunworks collector at the VPI & SU Swine Center.

pump 0-18.9 LPM, (0-5 GPM) was used. The water pump was connected to a 189.25 liter (50 gal) storage tank which was open to the atmosphere. Flowmeters and needle valve combinations were installed to measure and control flow through the system. Each collector had an individual flowmeter and the flow was varied between the collectors to test their efficiency and optimum flow rate.

#### WASTE TREATMENT LAGOON

The lagoon had surface dimensions of 30.5 m (100 ft) by 30.5 m (100 ft). The depth ranged from 0.9-1.5 m (3-5 ft) with an average volume of 1132.7 m<sup>3</sup> (40,000 ft<sup>3</sup>). Although there was heat stored in the lagoon fluid, it was difficult to remove when needed because the lagoon temperature varied between 2.5-7.2<sup>o</sup> C (36.5-45.0<sup>o</sup> F) during the winter months (Hill, 1975). The heat pump cannot function with input fluid temperature that low due to built-in protection to prevent freezing of the fluid which could damage the unit. There was a danger of freezing due to the possible 5.6<sup>o</sup> C (10<sup>o</sup> F) temperature drop in the working fluid as it passed through the water-to-refrigerant heat exchanger.

#### LAGOON HEAT EXCHANGER

A fresh water energy removal system was needed because of the corrosiveness of the lagoon water. Therefore, a lagoon heat exchanger was designed to remove heat from the lagoon in a way that would protect the heat pump. The

exchanger, Figure 7, consisted of 91.4 m (300 ft) of black plastic pipe, 1.27 cm (1/2 inch) nominal diameter with a wall thickness of 0.16 cm (1/16 inch). The pipe was formed into 50.8-60.96 cm (20-24 inch) diameter coils. Each coil was spaced 22.9 cm (9 inches) apart on a 5.1 cm x 5.1 cm (2 x 2) frame. The entire frame was submerged in the lagoon and anchored in place by concrete weights.

#### SYSTEM OPERATION

Two distinct modes of operation were used. These modes, as explained further below, were:

1. Operation when sufficient solar insolation or energy from storage was available.
2. Operation when sufficient solar insolation was not available.

Mode 1, when sufficient solar insolation or energy from storage was available, followed the flow pattern diagramed in Figure 8. The water was pumped from the storage tank through the heat exchanger in the lagoon. If the water flowing through the exchanger was cooled, valves were closed and the exchanger bypassed. If energy was gained, flow was continued through the exchanger and into the collectors. The collectors heated the water which then flowed to the heat pump. When circulating through the heat pump, energy was removed from the water to heat the building, as necessary. The water then returned to the storage

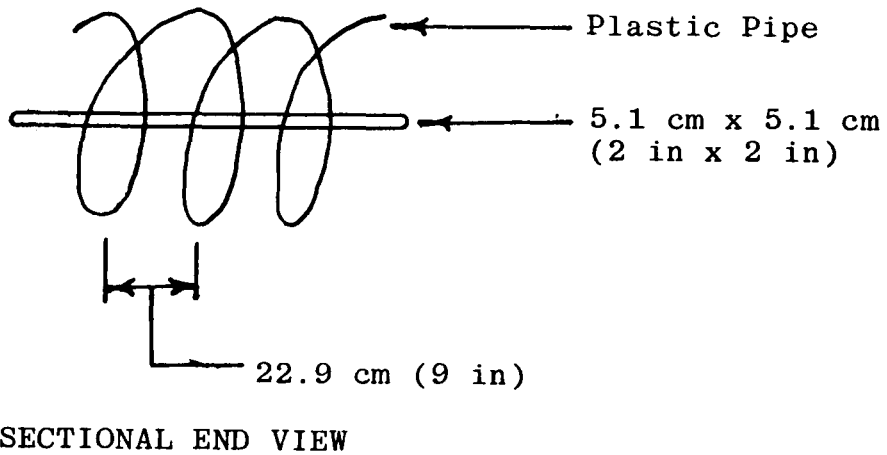
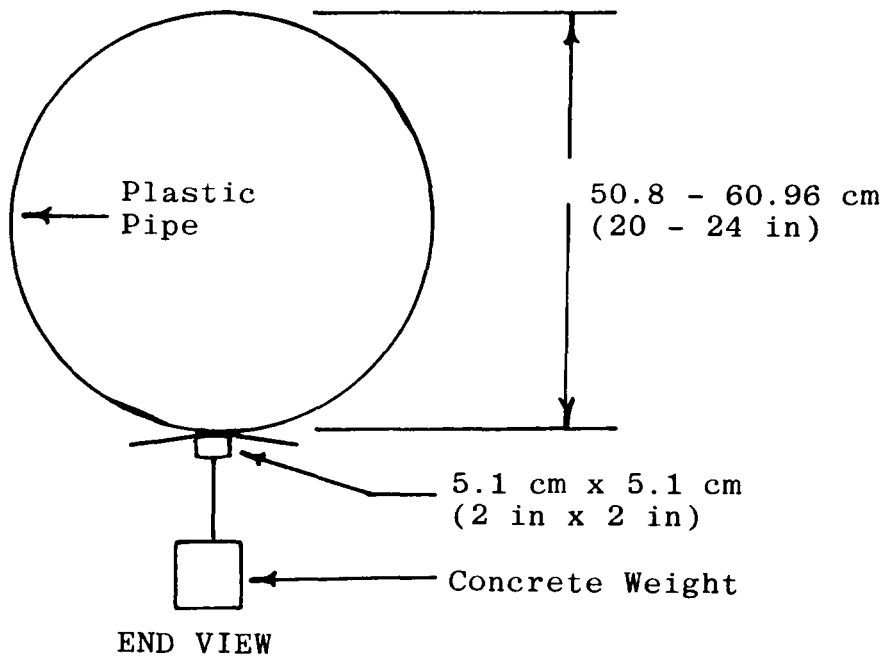


Figure 7. Schematic diagram of the lagoon heat exchanger.

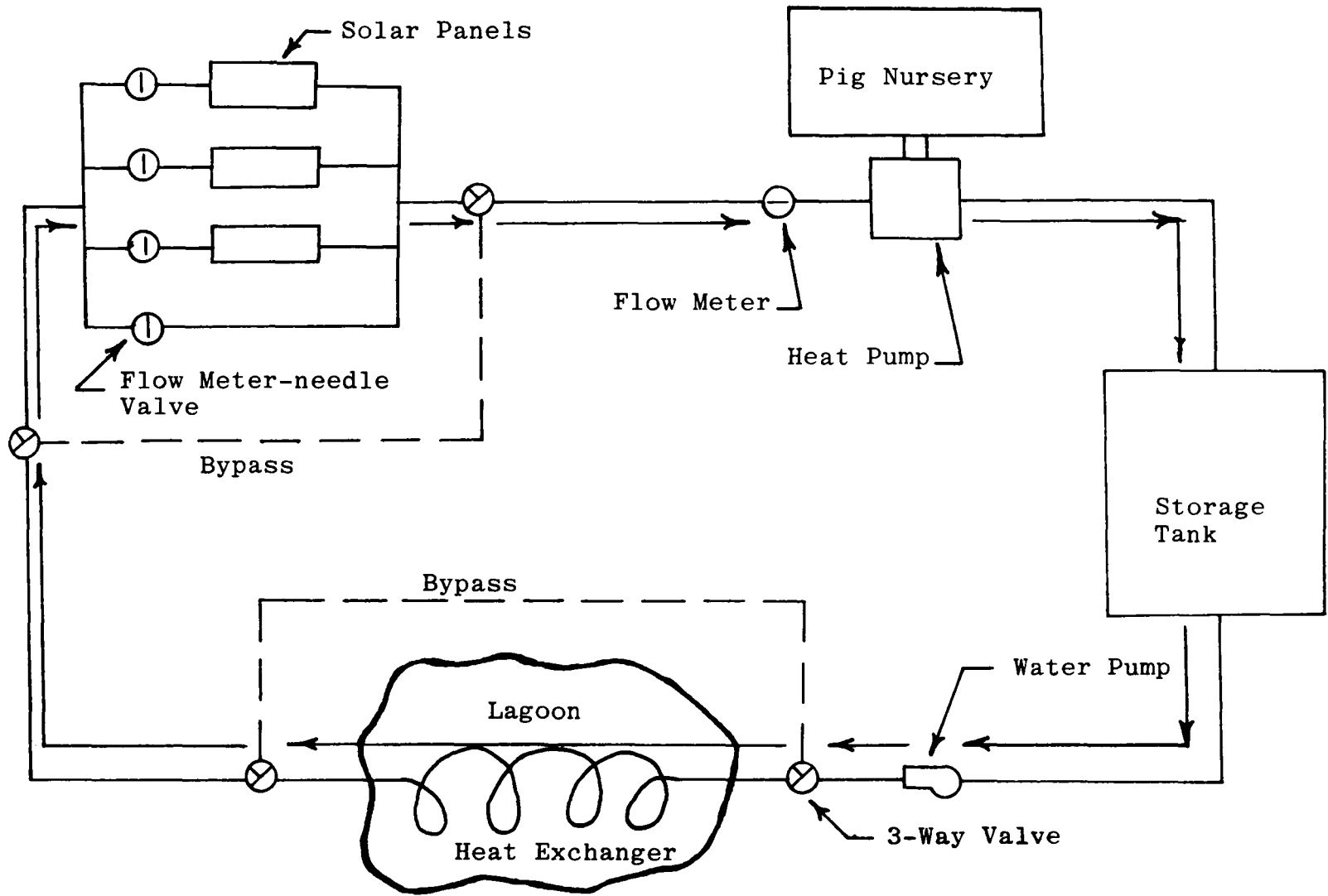


Figure 8. Schematic flow diagram of Mode 1.

tank for recirculation through the system. After sunset, the system would heat the building by removing energy in the form of heat from the storage tank. The building was heated by this process until the water temperature dropped below that of the lagoon. At that time, the flow pattern in Mode 2 became operational.

Mode 2, when sufficient solar insolation was not available, followed the flow pattern diagramed in Figure 9. When the temperature of the water in the storage tank was less than that of the lagoon, the water was pumped through the exchanger in the lagoon. As the water passed through the exchanger, energy was gained and the water warmed. The warmed water was then pumped through the heat pump and energy removed to heat the building, as necessary. The water then returned to the storage tank for recirculation through the system.

If the water did not gain enough energy to raise the temperature to a minimum of  $7.2^{\circ}\text{C}$  ( $45^{\circ}\text{F}$ ), the heat pump could not operate due to the built-in protection setting mentioned earlier. With this occurrence, the auxiliary heating system became operational and the water in the solar heating system was allowed to circulate for maximum heating. The solar assisted heat pump system became operational again when the water in the storage tank reached a temperature of  $7.2^{\circ}\text{C}$  ( $45^{\circ}\text{F}$ ).

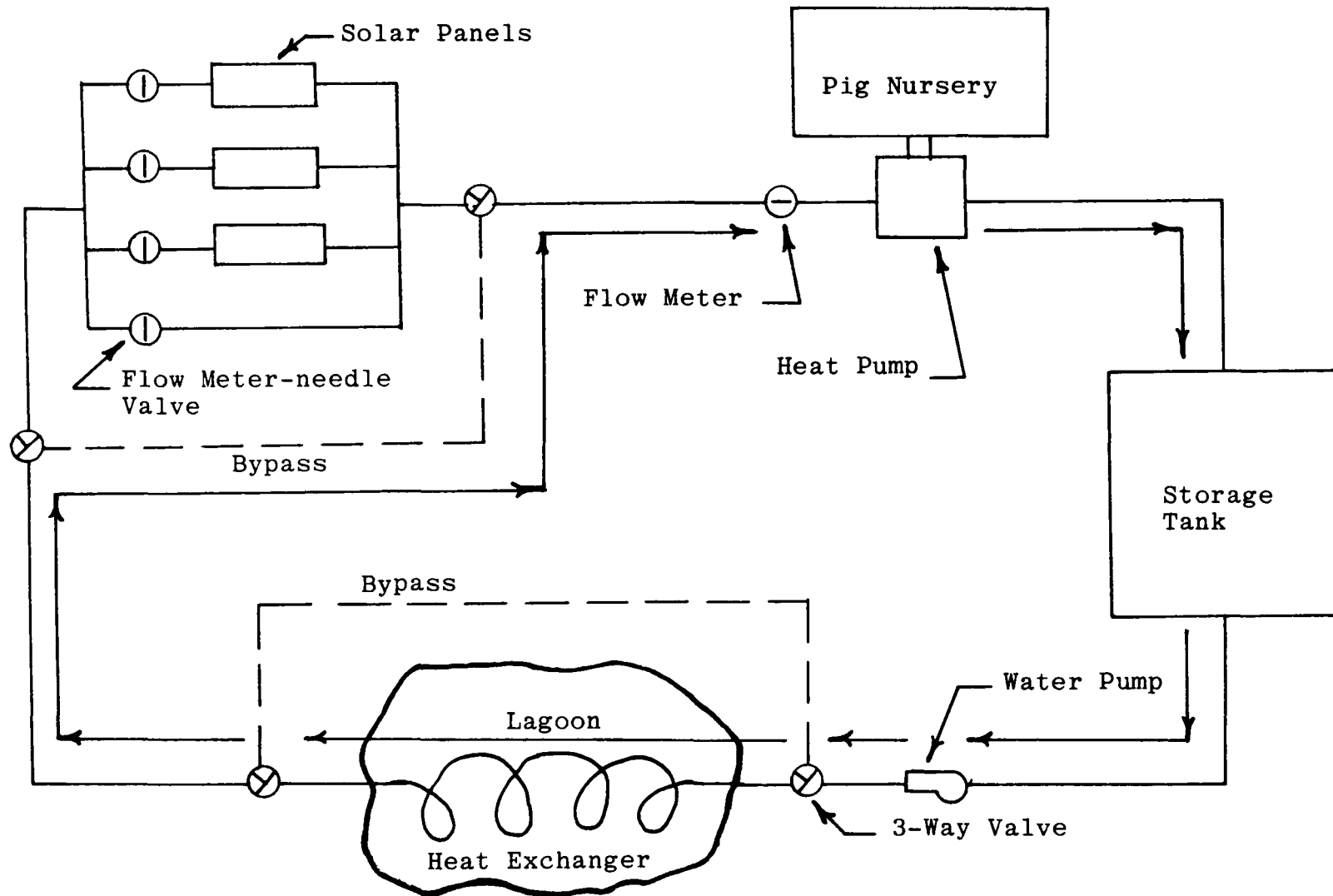


Figure 9. Schematic flow diagram of Mode 2.

## DATA COLLECTION

To record the performance of the system and evaluate the energy available in the lagoon, thermocouples were placed in selected areas of the flow pattern. These locations were:

1. Inlet and outlet of the heat exchanger in the lagoon;
2. Inlet and outlet of each collector;
3. Mixed outlet of the collectors;
4. Air inlet and outlet of the heat pump;
5. Water inlet and outlet of the heat pump;
6. Inside building temperature;
7. Inside water tank temperature;
8. Outside air temperature;
9. Lagoon temperature near the heat exchanger.

To securely mount each thermocouple, a hole slightly larger than the diameter of the thermocouple was drilled into pieces of plastic pipe. Each thermocouple was then properly placed in the water flow and glued with epoxy.

### Model Development

A computer model was developed to predict the amount of low level energy available in the waste treatment lagoon and to simulate the response of the lagoon to natural energy gains and losses as well as energy removal by the heat exchanger. The model also calculated the amount of

available solar insolation on a daily and hourly basis. By utilization of the computer model, the thermal stability of the waste treatment lagoon was evaluated.

#### AVAILABLE SOLAR INSOLATION

A combination of sine functions (Duffie and Beckman, 1974) were used to estimate the amount of available solar insolation. The first sine function, Figure 10, was used to estimate the total daily insolation. These equations were:

for  $I = 1-274$ ;

$$Q = 9000.0 \sin (0.0086 I + 0.785) + 21000.0 \quad (1a)$$

for  $I = 275-365$ ;

$$Q = 9000.0 \sin (0.0086 I + 3.925) + 21000.0 \quad (1b)$$

where:  $Q =$  solar insolation rate, watts/m<sup>2</sup> - day

$I =$  the number corresponding to the day of the year (March 26 = Day 1)

Equation 1a had a phase shift of  $\text{Pi}/4$  since the base function given by Duffie and Beckman (1974) was modified to begin on the spring equinox, March 26. A phase shift of  $1.25 \text{ Pi}$  was used in equation 1b to compensate for increased solar insolation after the shortest day of the year, December 26. The  $1.25 \text{ Pi}$  phase shift was necessary since the insolation rate increased significantly more than allowed for by the  $\text{Pi}/4$  phase shift.

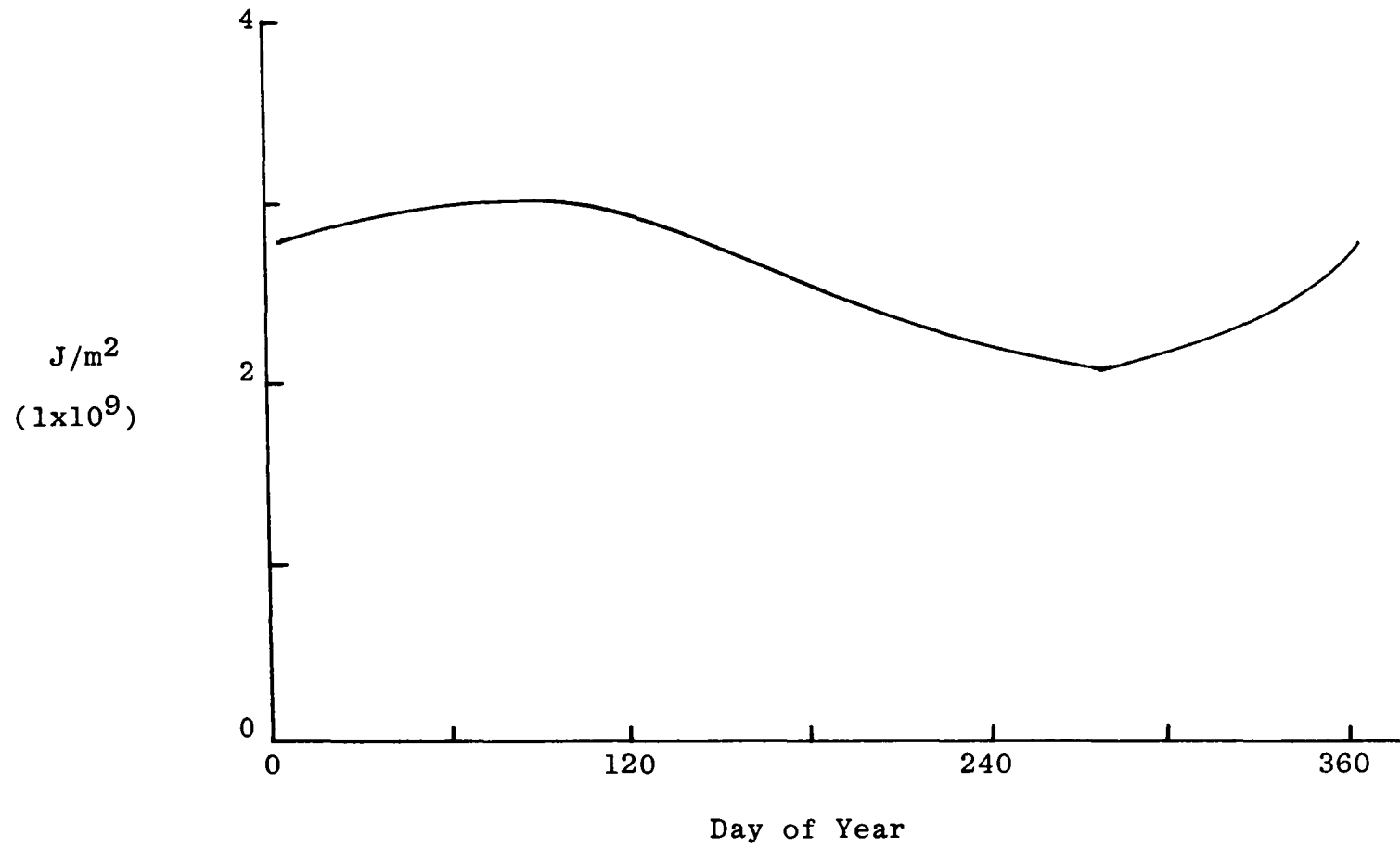


Figure 10. Daily solar insolation for clear conditions for the Roanoke area.

The second sine function, Figure 11, was used to determine the hourly distribution of solar energy during a given day. The function originally represented one particular day, March 26, and was modified to compensate for changes in day length and solar insolation.

To account for changes in day length, actual data for sunrise (DAWN) and sunset (DUSK) for Roanoke, Virginia was plotted. By fitting a series of three straight lines to data for both DAWN, Figure 12, and DUSK, Figure 13, changes in day length could be determined. These equations were:

$$\text{for } I = 1-86; \quad \text{DAWN } (I) = 375.0 - 0.884 I \quad (2a)$$

$$\text{DUSK } (I) = 1117.0 + 0.767 I \quad (2b)$$

$$\text{for } I = 87-280; \quad \text{DAWN } (I) = 230.0 + 0.799 I \quad (2c)$$

$$\text{DUSK } (I) = 1243.0 - 0.773 I \quad (2d)$$

$$\text{for } I = 281-365; \quad \text{DAWN } (I) = 715.0 - 0.929 I \quad (2e)$$

$$\text{DUSK } (I) = 752.0 + 0.988 I \quad (2f)$$

where: DAWN (I) = sunrise, minutes past 12:00 p.m.

DUSK (I) = sunset, minutes past 12:00 p.m.

I = the number corresponding to the  
day of the year (March 26 = Day 1).

By utilizing the day length as calculated from equations 2a-f, a variable constant for the hourly insolation rate was determined. Since the total available insolation for the referenced day, March 26, and any given day were

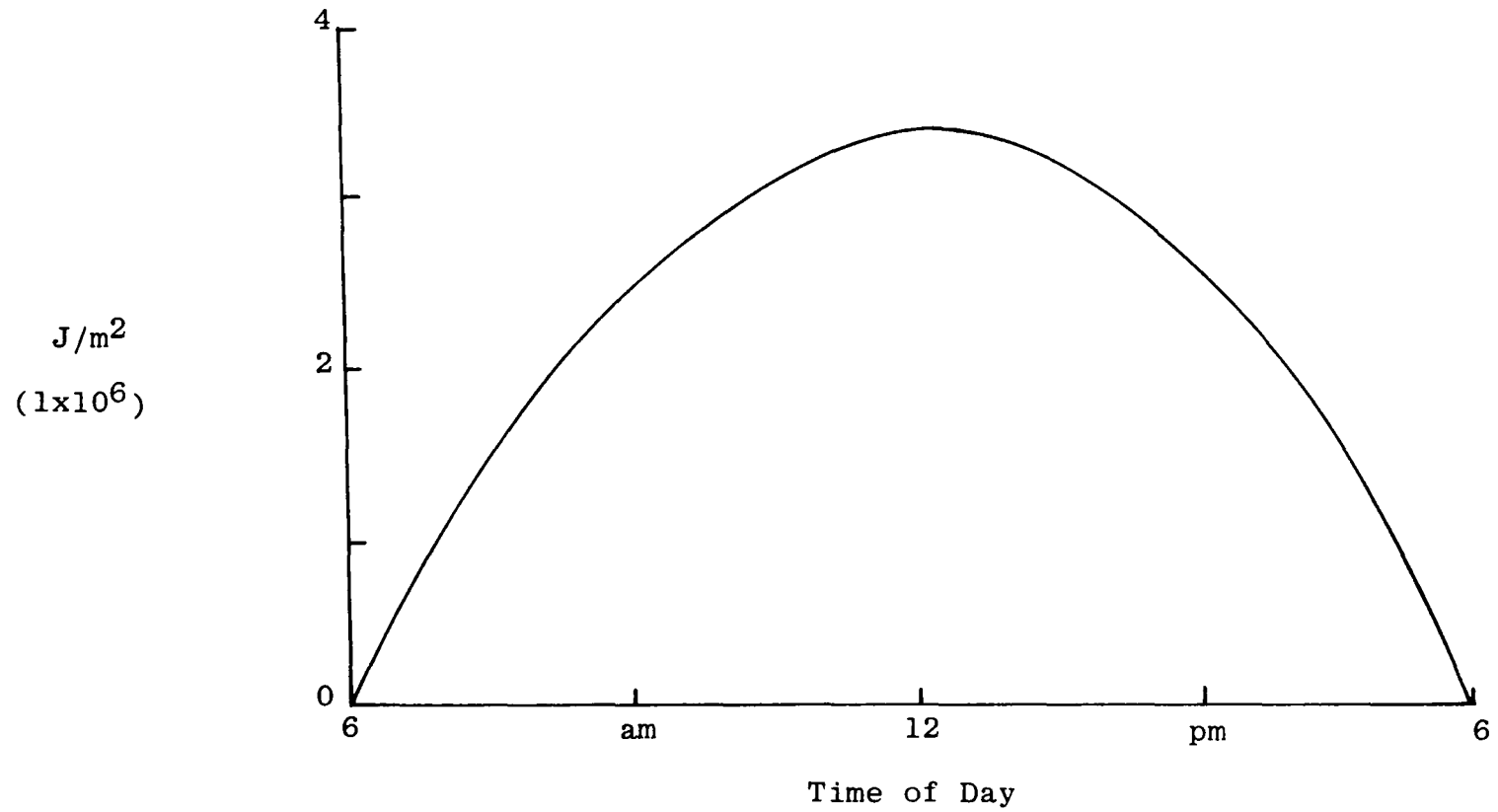


Figure 11. Hourly solar insolation on March 26 for the Roanoke area.

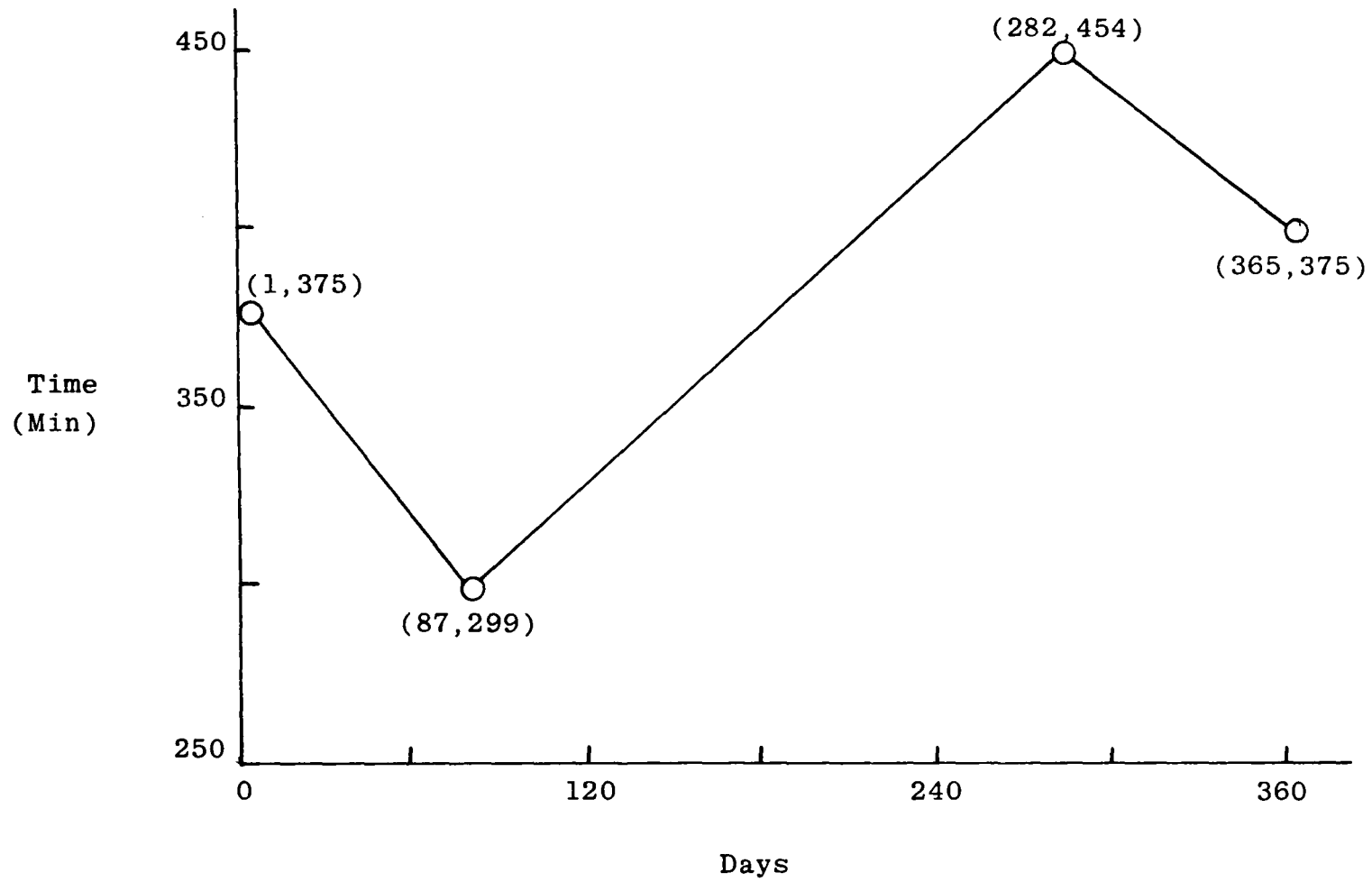


Figure 12. Sunrise for Roanoke, Virginia (minutes past midnight).

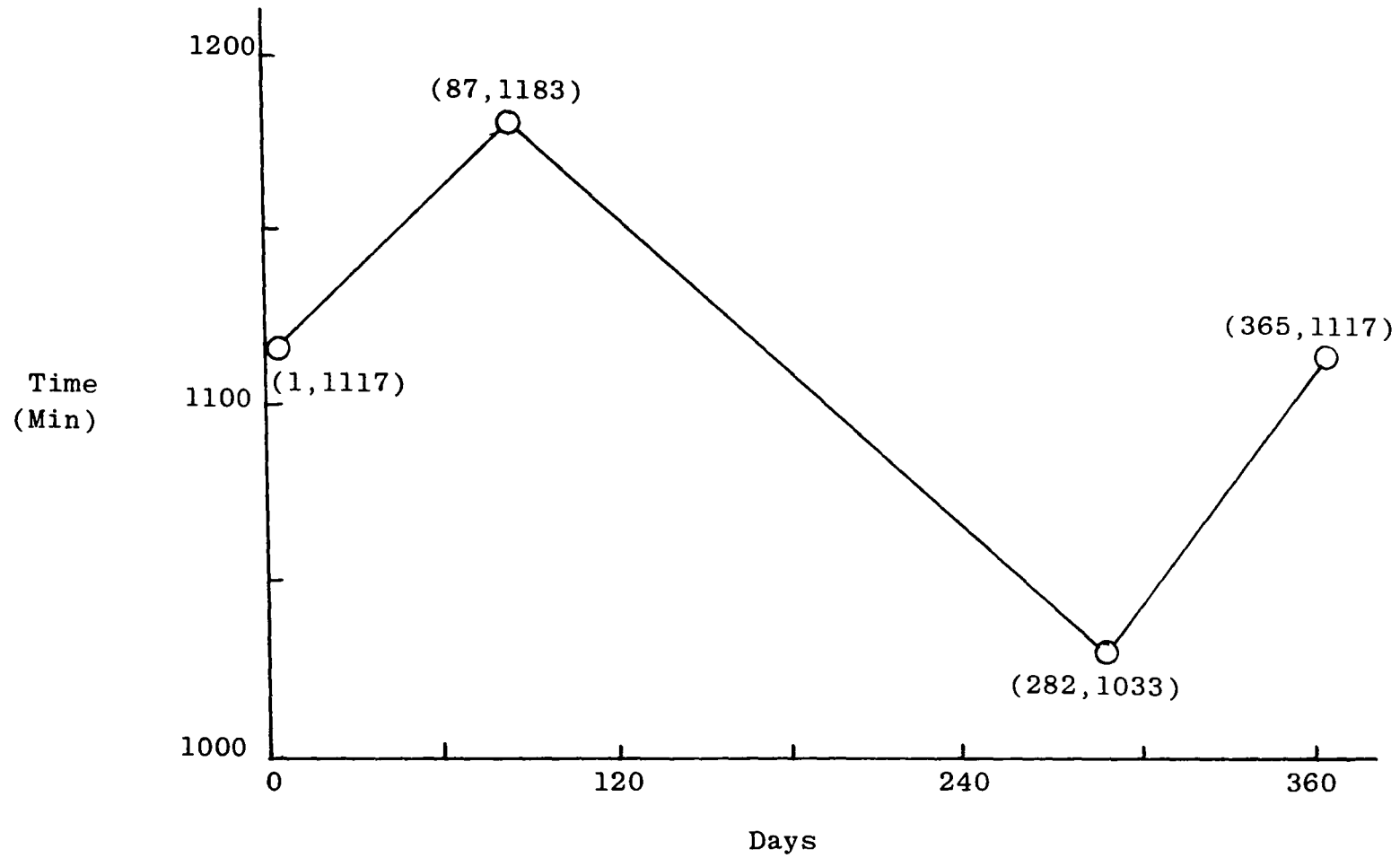


Figure 13. Sunset at Roanoke, Virginia (minutes past midnight).

known, both days were divided into 1000 intervals. By calculating the relationship for corresponding intervals between the two days, the hourly insolation curve for the given day was determined. The equation was

$$\text{DAY} = \text{CONST} (\sin (3.14 \text{ XM/DD}) ) \quad (3)$$

where: DAY = hourly insolation rate, J/m<sup>2</sup>

CONST = constant for the given day

XM = time past sunrise, minutes

DD = day length, minutes

The equations described above were used to calculate the insolation rates for clear days. Sample values are given in Table I.

#### LAGOON HEAT BALANCE

To model the response of the lagoon to natural conditions, an energy balance was developed for the lagoon. The energy lost by the lagoon was considered to occur by evaporation, radiation, convection and conduction. Under many normal weather conditions, the losses due to these factors may become gains when the lagoon temperature is less than the temperature of the surrounding earth and air. In this project, solar insolation was used as the only energy gain by the lagoon under the majority of weather conditions.

#### Energy Gain by Solar Energy

Using the equations developed above, the daily insola-

TABLE I. Available solar insolation for clear days as predicted  
by the computer model. ( $1 \times 10^6 \text{ J/m}^2$ )\*

Date	Time of Day (Selected Hourly Intervals)							Daily Total
	<u>5 - 6 (am)</u>	<u>7 - 8</u>	<u>9 - 10</u>	<u>11 - 12</u>	<u>1 - 2</u>	<u>3 - 4</u>	<u>5 - 6</u>	
Jan 1	0.00	0.12	2.09	3.36	3.25	1.81	0.01	21.53
Feb 1	0.00	0.45	2.32	3.40	3.32	2.12	0.25	23.86
March 1	0.00	0.82	2.47	3.40	3.35	2.32	0.62	25.80
April 1	0.00	1.16	2.57	3.36	3.32	2.47	1.01	27.62
May 1	0.04	1.39	2.64	3.31	3.27	2.52	1.23	28.92
June 1	0.21	1.57	2.64	3.21	3.17	2.51	1.38	29.70
July 1	0.27	1.62	2.68	3.22	3.16	2.49	1.36	29.87
August 1	0.08	1.50	2.71	3.36	3.28	2.51	1.21	29.42
Sep 1	0.001	1.31	2.70	3.43	3.35	2.46	0.98	28.36
Oct 1	0.00	1.06	2.61	3.45	3.36	2.35	0.70	26.84
Nov 1	0.00	0.72	2.45	3.42	3.31	2.17	0.37	24.86
Dec 1	0.00	0.36	2.25	3.34	3.22	1.95	0.12	22.68

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\*To convert from  $\text{J/m}^2$  to  $\text{BTU/ft}^2$ , divide by  $1.14 \times 10^4$

tion was calculated. To model actual conditions, a cloud factor for the Roanoke area was calculated from Weather Bureau data. The cloud factor was determined by calculating the ratio of measured versus predicted insolation. Equation 4 shows the determination of insolation while Equation 5 relates the insolation to the heat added to the lagoon.

$$\text{SOL} = \text{CF} \times (\text{QHD}(\text{N}, \text{II}) / 60.0) \quad (4)$$

where: SOL = actual insolation,  $\text{J}/\text{m}^2\text{-min}$

CF = cloud factor, 0.4

QHD(N,II) = hourly insolation for a clear day,  
 $\text{J}/\text{m}^2\text{-hr}$

N = the number corresponding to the day  
of the year (March 26 = Day 1)

II = hour of the day

$$\text{ADDHL} = 0.9 \times \text{SOL} \quad (5)$$

where: ADDHL = added heat to the lagoon,  $\text{J}/\text{m}^2\text{-min}$

SOL = computed insolation from Equation 4,  
 $\text{J}/\text{m}^2\text{-min}$

0.9 = absorptivity of water surface

While the gains by the lagoon were limited to solar insolation, heat losses included: 1.) conductive loss to the

ground, 2.) convective loss to the air, 3.) radiation loss and 4.) evaporation loss.

### Conductive Loss

Conductive loss to the ground may be as high as 10-25% in some instances (Usmanov, Umarov and Zakhidov, 1969). Although the loss can be high, the rate of loss decreases as the soil becomes heated to the lagoon temperature. The earth supplies energy to the lagoon when the water temperature is cooler than the earth temperature. Equation 6 was used to estimate conductive loss (Root, 1959).

$$Q_{\text{cond}} = h_c A \Delta t \quad (6)$$

where:  $Q_{\text{cond}}$  = conductive loss, J/hr

$h_c$  = thermal conductivity, J/hr-m<sup>2</sup>-°C

$A$  = area of the lagoon, m<sup>2</sup>

$\Delta t$  = temperature difference,

$T_{\text{water}} - T_{\text{ground}}, ^\circ\text{C}$

To simplify this calculation, the earth's temperature was assumed to be a constant 12.2° C (54° F) and the thermal conductivity of the soil to be 0.0325.

### Convective Loss

As with conductive loss, energy gain by convection is possible. Equation 7 was used to determine convective loss (Root, 1959).

$$Q_{\text{conv}} = h_{\text{cv}} A \Delta t \quad (7)$$

where:  $Q_{\text{conv}}$  = convective loss, J/hr

$h_{\text{cv}}$  = thermal conductivity of air,  
J/hr-m<sup>2</sup>-°C

$A$  = area of the lagoon, m<sup>2</sup>

$\Delta t$  = temperature difference,

$T_{\text{surface}} - T_{\text{air}}, ^\circ\text{C}$

To simplify this calculation, the wind velocity was assumed to be 1.2 kph (0.75 mph) and the average daily air temperature was calculated using Equation 8.

$$TA(\text{IIII}) = 20.2 \sin(0.017 \text{ IIII} + 50.0) \quad (8)$$

where:  $TA$  = average daily air temperature, °C

$\text{IIII}$  = the number corresponding to the day  
of the year (March 26 = Day 1)

The equation for average daily temperature was determined by fitting the proper sine function to collected data for the Blacksburg area.

### Radiation Loss

Equation 9 was used to determine the radiation loss to the atmosphere (Root, 1959).

$$Q_{\text{r}} = 0.173 A \varepsilon ((t_{\text{w}}/100)^4 - (t_{\text{s}}/100)^4) \quad (9)$$

where:  $Q_{\text{r}}$  = radiation loss, J/hr

$A$  = area of the lagoon, m<sup>2</sup>

$$\begin{aligned}\epsilon &= \text{emmissivity, J/hr-m}^2\text{-}^\circ\text{C} \\ t_w &= \text{lagoon temperature, }^\circ\text{C (}^\circ\text{R)} \\ t_s &= \text{air temperature, }^\circ\text{C (}^\circ\text{R)}\end{aligned}$$

To simplify the above calculation, the emissivity was assumed to be 0.95 and the average air temperature calculated using Equation 8.

### Evaporative Loss

Losses due to evaporation, in some instances, may be quite large. Equation 10 was used to determine the evaporative loss (Root, 1959).

$$Q_e = 1.08 (\mu \rho) \Delta p L_w A \quad (10)$$

where:  $Q_e$  = evaporative loss, J/hr

$\mu$  = air velocity, m/sec

$\rho$  = air density, g/m<sup>3</sup>

$\Delta p$  = pressure difference between the vapor pressure of water at wet-bulb air temperature and the partial pressure of water vapor in the air, g/cm<sup>2</sup>

$L_w$  = latent heat of steam at the wet-bulb air temperature, J/g

$A$  = area of the lagoon, m<sup>2</sup>

Due to time limitations and the complexity of modeling

the psychrometric chart, several simplifying assumptions were made in calculating evaporative losses. The simplifying assumptions were:

- 1.) air velocity = 1.21 km/hr (0.75 mph)
- 2.) air density = 12.02 kg/m<sup>3</sup> (0.75 lb/ft<sup>3</sup>)
- 3.)  $\Delta p = 0.14 \text{ g/cm}^3$  (0.002 psi)
- 4.) latent heat of steam = 2254.1 J/g (970 BTU/lb)

By utilizing these assumptions, the loss due to evaporation remained constant at  $3.0 \times 10^6$  J/hr (2846 BTU/hr) over the entire day. Since the evaporative loss remains constant, the computer model will have a slight built-in error which should not exceed 1<sup>o</sup> C (1.8<sup>o</sup> F) due to the volume of the waste treatment lagoon.

#### LAGOON HEAT EXCHANGER ENERGY BALANCE

The lagoon heat exchanger allows for the transfer of heat between the working medium, a water-antifreeze solution, and the waste treatment lagoon. To determine the energy transferred between the working medium and the waste treatment lagoon, a computer model was developed. The model predicted the energy transferred through the exchanger and the corresponding temperature changes in the working medium and lagoon temperatures.

To simplify the model, the outside and inside film coefficients were assumed to be constant over the length of

the exchanger. The outside film coefficient,  $f_o$ , was assumed to have a value of 6.0 (Baldwin, 1977). The inside film coefficient,  $f_i$ , was determined using the following procedure and equations. The kinematic viscosity and thermal conductivity of the working medium were required to compute  $f_i$ . The equations used to calculate these values were (Holman, 1963):

$$\nu_m = 4.56 \times 10^{-4} / t_m^{0.90458} \quad (11)$$

$$k_m = 2.8175 t_m^{0.9551} \quad (12)$$

where:  $\nu_m$  = kinematic viscosity of the working medium,  $m^2/sec$

$k_m$  = thermal conductivity of the working medium,  $J/hr m^2 \text{ } ^\circ C$

$t_m$  = temperature of the working medium,  $^\circ C$

Given the kinematic viscosity, the Reynolds Number to the entrance of the exchanger was calculated as shown in Equations 13 and 14 (Holman, 1963).

$$R = 2 r_i V / \nu_m \quad (13)$$

$$V = \dot{V}_m / \pi r_i^2 \quad (14)$$

where:  $R$  = Reynolds Number

$r_i$  = inside radius of the tubing, m

$V$  = average velocity of the medium, m/sec

$\nu_m$  = kinematic viscosity of the working medium,  
 $m^2/sec$

$\dot{V}_m$  = volumetric flow rate of the working medium,  
 $m^3/sec$

Combining Equations 13 and 14 yielded Equation 15.

$$R = 2 \dot{V}_m / \pi r_i \nu_m \quad (15)$$

The Prandtl Number, which gives an indication of shock phenomena in fluid flow, was calculated using  $\nu_m$  and  $k_m$  for the entrance to the lagoon heat exchanger. This equation was (Holman, 1963):

$$Pr = \nu_m \rho c_p / k_m \quad (16)$$

where: Pr = Prandtl Number

$\nu_m$  = kinematic viscosity,  $m^2/sec$

$k_m$  = thermal conductivity of the working medium,  
 $J/hr-m^2-^{\circ}C$

$\rho$  = density of the working medium,  $g/m^3$

$c_p$  = specific heat of the working medium,  
 $J/g-^{\circ}C$

The Nusselt Number for the entrance to the lagoon exchanger was calculated from Equation 17 (Holman, 1963).

$$\text{Nu} = 0.023 R^{0.8} \text{Pr}^n \quad (17)$$

where: Nu = Nusselt Number

R = Reynolds Number

Pr = Prandtl Number

$$n = \begin{cases} 0.4 & \text{when } t_m \leq t_2 \\ 0.3 & \text{when } t_m > t_2 \end{cases}$$

where:  $t_m$  = temperature of the working medium  
entering the exchanger, °C

$t_2$  = lagoon temperature, °C

After R, Pr, Nu,  $k_m$  and  $v_m$  were known, the inner film coefficient was calculated using Equation 18 (Holman, 1963).

$$f_i = \text{Nu } k_m / 2 r_i \quad (18)$$

where:  $f_i$  = inner film coefficient, J/hr-m-°C

Nu = Nusselt Number

$k_m$  = thermal conductivity of the working  
medium, J/hr-m<sup>2</sup>-°C

$r_i$  = inside radius of the exchanger tubing, m

Two additional parameters, 1.) the overall heat transfer coefficient of the exchanger tubing and 2.) the mass flow rate of the working medium, were required before the energy balance for the heat exchanger could be modeled. The

exchanger heat transfer coefficient and the mass flow rate were given by Equations 19 and 20.

$$U = \frac{1}{\frac{1}{f_o} + \frac{1}{k_e} + \frac{1}{f_i}} \quad (19)$$

where:  $U$  = heat transfer coefficient, J/hr-m-°C

$f_o$  = outside film coefficient, J/hr-m-°C

$f_i$  = inside film coefficient, J/hr-m-°C

$k_e$  = thermal conductivity of the exchanger tubing, J/hr-m-°C

$$\dot{m} = \rho \dot{V}_m \quad (20)$$

where:  $\dot{m}$  = mass flow rate, g/hr

$\rho$  = density of the medium, g/m<sup>3</sup>

$\dot{V}_m$  = volumetric flow rate of the medium,  
m<sup>3</sup>/hr

The exchanger energy balance per unit length of tubing was modeled from these equations. Differential Equation 21 for the temperature change along the tubing was used to determine the energy transferred between the working medium and the lagoon (Vaughan, 1976).

$$\frac{dt_m}{dx} = \frac{2 \pi r_o U (t_m - t_2)}{\dot{m} c_p} \quad (21)$$

The solution of the above differential equation for the entire length of tubing,  $L$ , is given in Equation 22 (Vaughan, 1976).

$$\frac{t_1 - t_2}{t_m - t_2} = e^{\frac{-2 \pi r_o L U}{\dot{m} c_p}} \quad (22)$$

where:  $t_1$  = temperature leaving the exchanger,  $^{\circ}\text{C}$   
 $t_2$  = temperature of the lagoon,  $^{\circ}\text{C}$   
 $t_m$  = temperature of the working medium,  $^{\circ}\text{C}$   
 $r_o$  = outside radius of the tubing, m  
 $L$  = length of exchanger tubing, m  
 $U$  = thermal conductivity of the exchanger tubing,  $\text{J/hr-m-}^{\circ}\text{C}$   
 $\dot{m}$  = mass flow rate, g/hr  
 $c_p$  = specific heat of the working medium,  $\text{j/g-}^{\circ}\text{C}$

The rate of energy transfer between the working medium and the lagoon is given by Equation 23.

$$Q = \dot{m} c_p ( t_m - t_2 ) \quad (23)$$

where:  $Q$  = energy transfer, J/hr  
 $\dot{m}$  = mass flow rate, g/hr  
 $c_p$  = specific heat of the working medium,  $\text{J/g-}^{\circ}\text{C}$

$t_m$  = working medium temperature, °C

$t_2$  = lagoon temperature, °C

By utilizing Equations 11 through 23, the energy transfer between the lagoon and the working medium was modeled. The modeling of this transfer of energy also served to test the feasibility of the heat exchanger as a means for removing energy from the lagoon. By evaluation of the computer model and collected data, a more efficient design of the heat exchanger may be developed.

#### OPERATIONALIZING THE COMPUTER MODEL

By summing the previously discussed heat flows, a composite model of the lagoon heat flow characteristics was constructed. After initial testing to assure computational accuracy, a comparison of model output to actual conditions was needed. After comparing the predicted output to collected data, modifications were needed due to large increases in the lagoon temperature. After manual evaluation of the model, changes in the quantity of solar insolation being absorbed by the lagoon were made in the cloud factor. The cloud factor was increased to decrease the amount of insolation incident on the lagoon. After modification, the model was tested and the output evaluated again. The modification in the cloud factor altered the predicted lagoon temperature within a range of  $\pm 10\%$  of the collected data. The

computer model was then assumed to be accurate and the final computer simulation completed. The complete computer model is included in Appendix A with a breakdown of computer days in Appendix B.

## RESULTS AND DISCUSSION

The computer simulation program and collected data were used to determine the feasibility of utilizing the waste treatment lagoon as an energy source. The results of this evaluation are presented in the following narrative.

Energy added to the lagoon by solar insolation, calculated using Equation 5, is shown in Table II. The added energy varied with day length. In early morning, the energy gained was low and increased to a maximum near midday. After midday, energy added to the lagoon decreased with the decrease in solar insolation and finally became nil after sunset.

Conductive loss from the lagoon, calculated using Equation 6 is given in Table III. In some instances, the loss actually became a gain due to the earth temperature being warmer than the lagoon temperature. From the data in Table III, the earth provided energy to the lagoon for part of the day until solar insolation raised the lagoon temperature above  $12.2^{\circ}\text{C}$  ( $54^{\circ}\text{F}$ ). At that time, the lagoon began to lose energy to the earth and continued to do so until the lagoon temperature dropped below the earth's temperature. In some instances, the earth supplied energy to the lagoon for the entire day (December 1). This phenomenon occurred because the energy gain by the lagoon could not balance the

TABLE II. Solar energy added to the waste treatment lagoon as predicted  
by the computer model. ( $1 \times 10^9$  J)\*

Date	Time of Day (Selected Hourly Intervals)						
	<u>6 - 7 (am)</u>	<u>8 - 9</u>	<u>10 - 11</u>	<u>12 - 1</u>	<u>2 - 3</u>	<u>4 - 5</u>	<u>6 - 7</u>
Oct 12	0.02	0.61	1.04	1.17	0.98	0.48	0.0 <sup>+</sup>
Oct 13	0.02	0.60	1.04	1.17	0.97	0.49	0.0 <sup>+</sup>
Oct 14	0.02	0.60	1.03	1.17	0.97	0.49	0.0 <sup>+</sup>
Oct 15	0.02	0.60	1.03	1.17	0.97	0.49	0.0 <sup>+</sup>
Oct 21	0.01	0.58	1.03	1.17	0.97	0.47	0.0
Oct 22	0.01	0.58	1.03	1.17	0.97	0.47	0.0
Oct 23	0.01	0.58	1.03	1.17	0.97	0.47	0.0
Oct 24	0.01	0.58	1.03	1.17	0.97	0.47	0.0
Nov 3	0.0 <sup>+</sup>	0.54	1.01	1.17	0.95	0.43	0.0
Nov 4	0.0 <sup>+</sup>	0.54	1.01	1.17	0.95	0.43	0.0
Nov 5	0.0 <sup>+</sup>	0.54	1.01	1.17	0.95	0.43	0.0
Nov 6	0.0 <sup>+</sup>	0.54	1.01	1.17	0.94	0.42	0.0
Dec 1	0.0	0.45	0.97	1.14	0.90	0.34	0.0
Dec 2	0.0	0.45	0.97	1.14	0.90	0.33	0.0
Dec 3	0.0	0.44	0.97	1.14	0.90	0.33	0.0
Dec 4	0.0	0.44	0.97	1.14	0.90	0.32	0.0

\*To convert from J to BTU, divide by 1055

+Energy is added but on a  $1 \times 10^6$  J scale or less

TABLE III. Conductive energy loss from the waste treatment lagoon as predicted by the computer model. ( $1 \times 10^6$  J)\*

Date	Time of Day (Selected Hourly Intervals)						
	<u>6 - 7 (am)</u>	<u>8 - 9</u>	<u>10 - 11</u>	<u>12 - 1</u>	<u>2 - 3</u>	<u>4 - 5</u>	<u>6 - 7</u>
Oct 12	0.08	0.07	0.16	0.31	0.45	0.52	0.48
Oct 13	-0.03 <sup>1</sup>	-0.05	0.04	0.19	0.33	0.40	0.36
Oct 14	-0.15	-0.17	-0.08	0.71	0.21	0.28	0.24
Oct 15	-0.27	-0.29	-0.19	-0.04	0.10	0.16	0.12
Nov 3	-2.39	-2.41	-2.32	-2.17	-2.03	-1.97	-2.02
Nov 4	-2.49	-2.52	-2.43	-2.28	-2.13	-2.07	-2.12
Nov 5	-2.59	-2.62	-2.53	-2.37	-2.23	-2.17	-2.22
Nov 6	-2.69	-2.72	-2.63	-2.48	-2.33	-2.27	-2.32
Dec 1	-4.73	-4.76	-4.68	-4.52	-4.38	-4.32	-4.37
Dec 2	-4.79	-4.83	-4.74	-4.58	-4.44	-4.38	-4.44
Dec 3	-4.85	-4.88	-4.80	-4.64	-4.50	-4.44	-4.49
Dec 4	-4.91	-4.94	-4.86	-4.70	-4.55	-4.50	-4.55

\*To convert from J to BTU, divide by 1055

<sup>1</sup>Negative values represent energy being supplied to the lagoon

losses and the lagoon temperature remained below  $12.2^{\circ}$  C ( $54^{\circ}$  F).

Convective loss from the lagoon, calculated using Equation 7 and shown in Table IV, remained fairly constant over the day length and, in this study, accounted for the major portion of the total heat loss. Convective loss increased during the day due to the warming of the lagoon by solar insolation. This trend continued until the absorbed insolation no longer warmed the lagoon and convective loss began to decrease. On days when the air temperature was warmer than the lagoon temperature, energy was gained by convection.

Radiation loss to the atmosphere, computed with Equation 9 and shown in Table V, was approximately 10% of the convective loss. Radiation loss increased during the day due to the warming of the lagoon by solar insolation. The rate of energy loss remained fairly constant,  $\pm 10\%$ , over the day length and began to decrease with the decrease of solar insolation near sunset.

Evaporative loss from the lagoon, calculated using Equation 10, remained constant for the entire computer simulation due to the simplifying assumptions. The rate of energy loss was determined to be  $3.0 \times 10^6$  J/hr (2846 BTU/hr) for each hour of the day. As shown in Table IV, the amount of energy lost due to evaporation was approximately 1% of

TABLE IV. Convective energy loss from the waste treatment lagoon as predicted by the computer program ( $1 \times 10^8$  J)\*

Date	Time of Day (Selected Hourly Intervals)						
	<u>6 - 7 (am)</u>	<u>8 - 9</u>	<u>10 - 11</u>	<u>12 - 1</u>	<u>2 - 3</u>	<u>4 - 5</u>	<u>6 - 7</u>
Oct 12	3.33	3.32	3.41	3.54	3.67	3.74	3.70
Oct 13	3.33	3.31	3.40	3.54	3.67	3.73	3.70
Oct 14	3.32	3.31	3.39	3.53	3.66	3.72	3.68
Oct 15	3.32	3.30	3.39	3.53	3.66	3.72	3.68
Nov 3	3.15	3.13	3.22	3.36	3.49	3.55	3.50
Nov 4	3.14	3.12	3.21	3.35	3.48	3.54	3.49
Nov 5	3.13	3.11	3.19	3.34	3.47	3.53	3.48
Nov 6	3.12	3.10	3.18	3.32	3.46	3.51	3.47
Dec 1	2.80	2.77	2.85	3.00	3.13	3.18	3.13
Dec 2	2.79	2.76	2.84	2.98	3.12	3.17	3.12
Dec 3	2.77	2.74	2.82	2.96	3.10	3.15	3.10
Dec 4	2.76	2.73	2.81	2.95	3.09	3.14	3.09

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\*To convert from J to BTU, divide by 1055

TABLE V. Radiation energy loss from the waste treatment lagoon as predicted by the computer model. ( $1 \times 10^7$  J)\*

Date	Time of Day (Selected Hourly Intervals)						
	<u>6 - 7 (am)</u>	<u>8 - 9</u>	<u>10 - 11</u>	<u>12 - 1</u>	<u>2 - 3</u>	<u>4 - 5</u>	<u>6 - 7</u>
Oct 12	5.00	4.98	5.11	5.33	5.53	5.63	5.56
Oct 13	4.98	4.96	5.09	5.31	5.51	5.61	5.54
Oct 14	4.96	4.93	5.07	5.28	5.49	5.58	5.52
Oct 15	4.94	4.91	5.05	5.26	5.47	5.56	5.50
Nov 3	4.49	4.46	4.58	4.79	4.99	5.08	5.01
Nov 4	4.47	4.44	4.56	4.77	4.97	5.05	4.98
Nov 5	4.44	4.41	4.53	4.74	4.94	5.02	4.95
Nov 6	4.42	4.39	4.51	4.72	4.91	5.00	4.93
Dec 1	3.81	3.76	3.88	4.08	4.27	4.35	4.27
Dec 2	3.78	3.74	3.85	4.05	4.24	4.32	4.25
Dec 3	3.76	3.71	3.82	4.03	4.22	4.29	4.22
Dec 4	3.73	3.69	3.80	4.01	4.20	4.27	4.20

\*To convert from J to BTU, divide by 1055

the convective loss.

Energy gain or loss by the working medium through the exchanger, calculated using Equations 11 through 23, is shown in Table VI. The working medium lost energy to the lagoon in the early morning, late afternoon and throughout the night. As the lagoon was warmed by solar insolation in the morning, energy was gained by the working medium from the lagoon. Since the working medium and lagoon had approximately the same temperatures (Figures 14 and 15), the total quantity of energy transferred was small when compared to the overall energy loss of the lagoon. Also, for purposes of this simulation, no energy was being removed from the working medium to heat the nursery and no temperature differential existed. To test the energy recovery of the working medium, energy was removed from the medium to heat the pig nursery under the system test procedure. As seen in Figure 16, the working medium gained energy from the lagoon and once again returned to thermal equality with the lagoon.

The overall energy balance for the lagoon was determined using Equations 1 through 23 and computed values are given in Table VII. Energy loss was high in the early morning and late afternoon when solar insolation was not available or in sufficient quantity to balance the losses. During most of the day when solar insolation was available, energy gains by the lagoon were quite large. Lagoon tempera-

TABLE VI. Lagoon exchanger energy flow to the working medium as predicted by the computer model. ( $1 \times 10^4$  J)\*

Date	Time of Day (Selected Hourly Intervals)						
	<u>6 - 7 (am)</u>	<u>8 - 9</u>	<u>10 - 11</u>	<u>12 - 1</u>	<u>2 - 3</u>	<u>4 - 5</u>	<u>6 - 7</u>
Oct 12	-6.14 <sup>1</sup>	0.74	8.37	11.85	10.07	3.58	-4.75
Oct 13	-6.14	0.70	8.36	11.86	10.08	3.58	-4.75
Oct 14	-6.14	0.65	8.34	11.86	10.09	3.56	-4.75
Oct 15	-6.12	0.68	8.38	11.88	10.06	3.48	-4.80
Nov 3	-5.83	0.11	8.26	12.10	10.19	3.18	-5.12
Nov 4	-5.81	0.15	8.30	12.11	10.17	3.10	-5.16
Nov 5	-5.78	0.11	8.28	12.12	10.18	3.10	-5.14
Nov 6	-5.76	0.07	8.27	12.13	10.19	3.08	-5.16
Dec 1	-5.12	-0.64	8.14	12.35	10.31	2.56	-4.99
Dec 2	-5.09	-0.68	8.12	12.44	10.32	2.56	-4.97
Dec 3	-5.06	-0.73	8.11	12.45	10.33	2.56	-4.95
Dec 4	-5.03	-0.68	8.15	12.46	10.30	2.46	-4.94

\*To convert from J to BTU, divide by 1055

<sup>1</sup>Negative values mean energy is being supplied to the lagoon

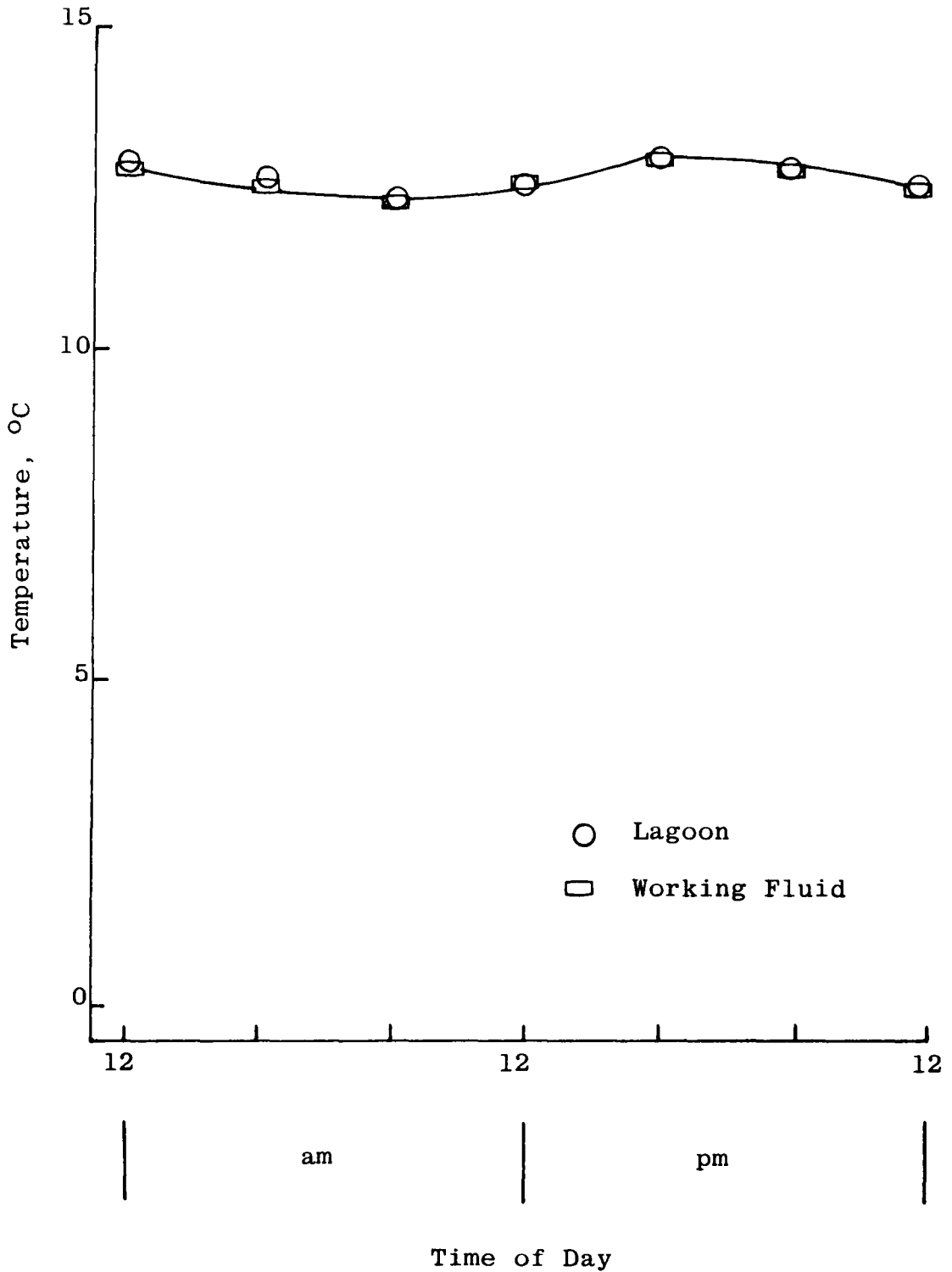


Figure 14. Comparison of predicted temperatures for the working fluid and lagoon (October 12).

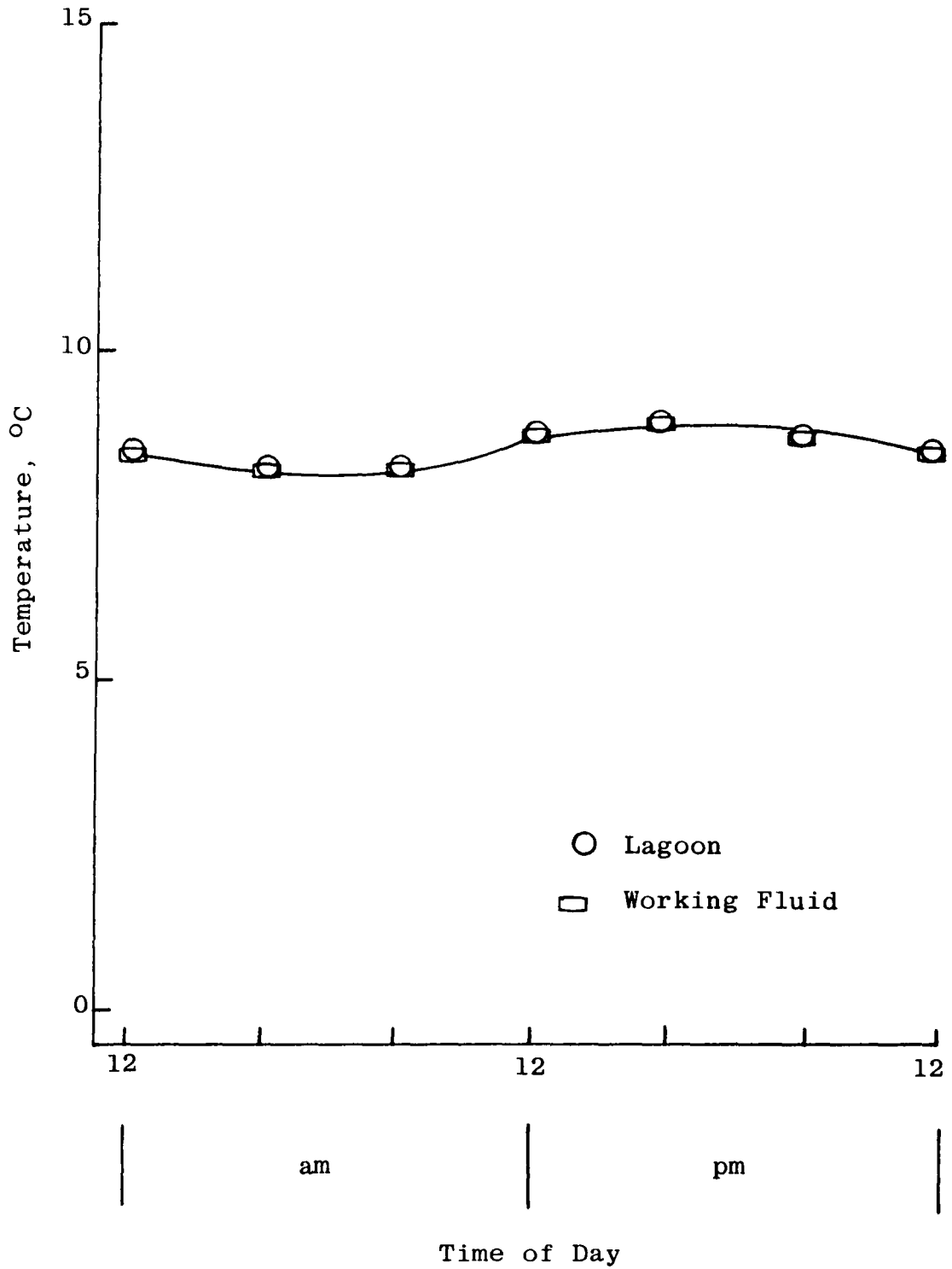


Figure 15. Comparison of predicted temperatures for the working fluid and lagoon (November 3).

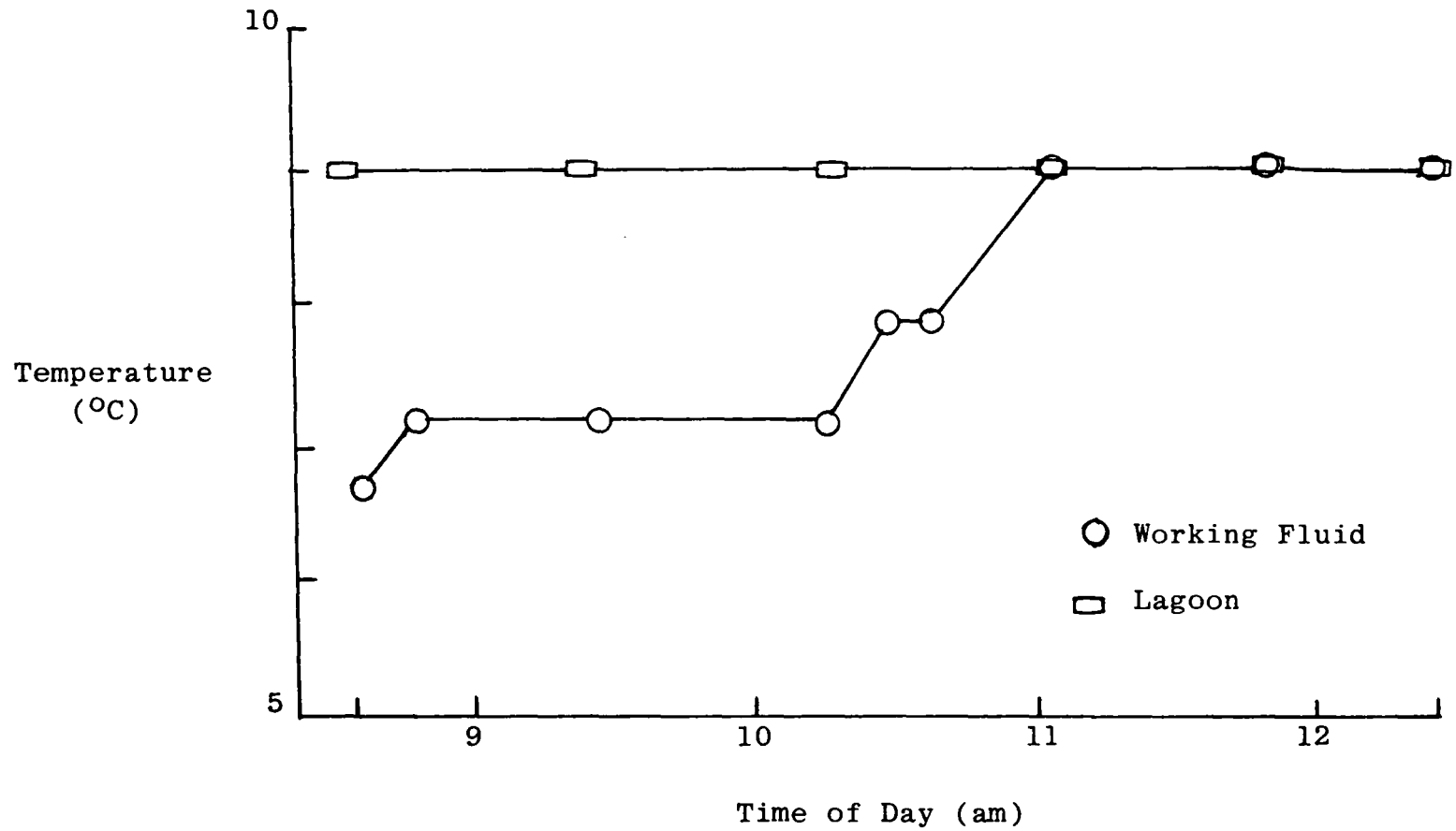


Figure 16. Recovery rate of the working fluid after energy removal (October 26).

TABLE VII. Overall energy loss from the waste treatment lagoon as predicted by the computer model. ( $1 \times 10^7$  J)\*

Date	Time of Day (Selected Hourly Intervals)						
	<u>6 - 7 (am)</u>	<u>8 - 9</u>	<u>10 - 11</u>	<u>12 - 1</u>	<u>2 - 3</u>	<u>4 - 5</u>	<u>6 - 7</u>
Oct 12	36.65	-22.06 <sup>1</sup>	-64.31	-76.31	-54.90	-6.44	42.84
Oct 13	36.76	-21.81	-64.27	-76.39	-54.95	-6.35	42.76
Oct 14	36.85	-21.55	-64.24	-76.47	-55.00	-6.27	42.66
Oct 15	36.75	-21.75	-64.43	-76.51	-54.75	-5.70	42.56
Nov 3	35.96	-18.47	-64.66	-78.18	-55.04	-2.81	40.10
Nov 4	35.82	-18.70	-64.89	-78.23	-54.77	-2.21	39.95
Nov 5	35.70	-18.45	-64.86	-78.33	-54.84	-2.14	39.80
Nov 6	35.59	-18.19	-64.83	-78.42	-54.91	-2.07	39.65
Dec 1	31.64	-14.04	-65.27	-80.62	-54.83	2.53	35.46
Dec 2	31.46	-13.75	-65.23	-80.72	-54.91	2.61	35.28
Dec 3	31.29	-13.47	-65.18	-80.82	-54.97	2.69	35.09
Dec 4	31.11	-13.71	-65.42	-80.86	-54.66	3.34	34.91

\*To convert from J to BTU, divide by 1055

<sup>1</sup>Negative values mean energy is being supplied to the lagoon

ture, predicted from the overall energy loss or gain by the lagoon (Figures 17 through 19), was slightly higher than the collected data with an error less than 10%, probably due to simplifications in earlier calculations. One simplification which may have caused error involved the computation of evaporative loss. In natural bodies, up to 77% of the total loss may be due to evaporation (Usmanov and Eliseev, 1973).

Although total energy loss or gain by the lagoon was quite large, as seen in Tables II through VII, the corresponding energy loss to temperature change was large (Table VIII). For example, a temperature change of  $+0.03^{\circ}\text{C}$  ( $+0.06^{\circ}\text{F}$ ) required an energy gain of  $2.2 \times 10^8 \text{ J}$  ( $2.09 \times 10^5 \text{ BTU}$ ). Thus, large amounts of energy are available in the lagoon.

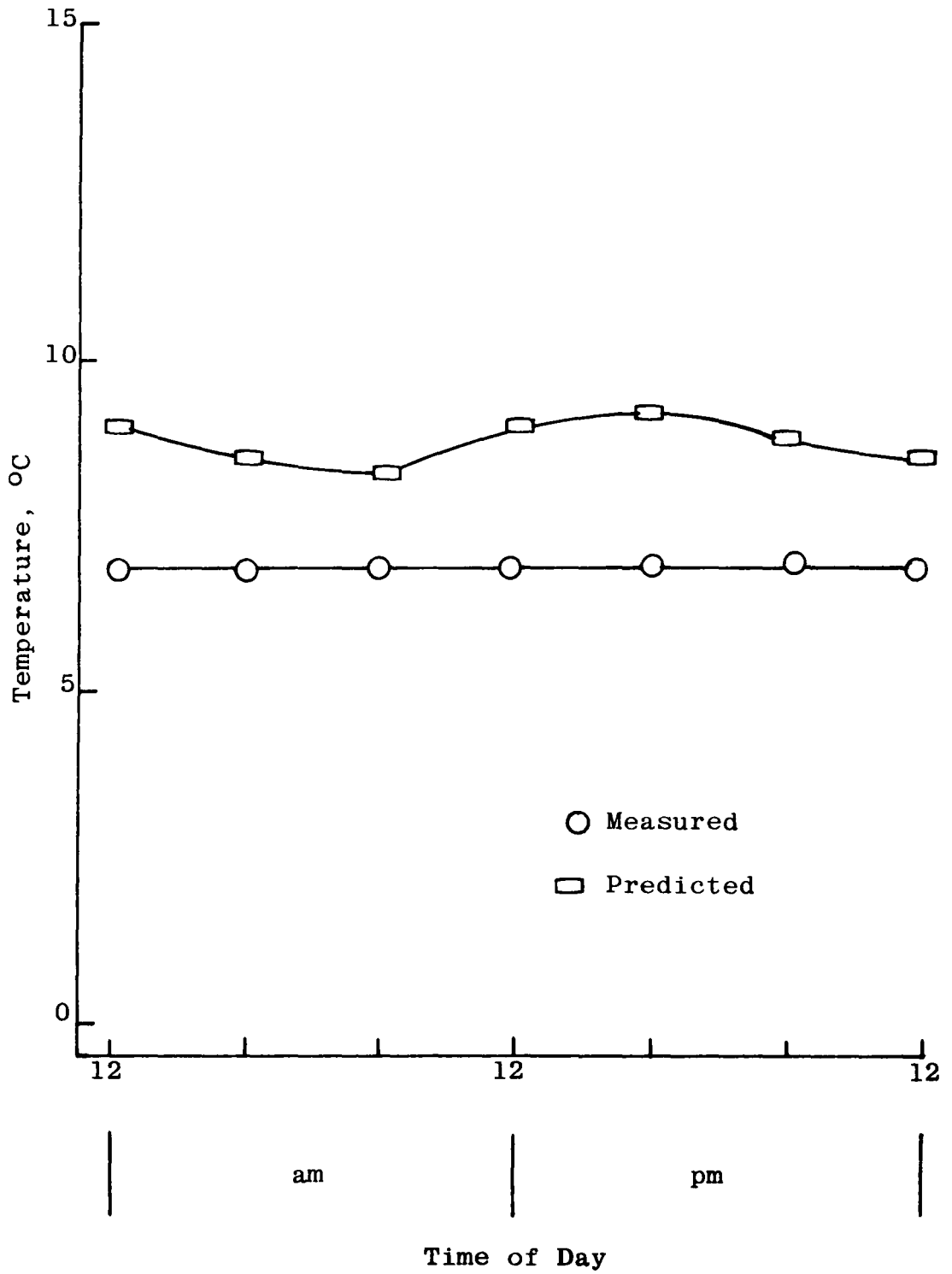


Figure 17. Comparison of predicted and measured waste treatment lagoon temperatures (November 3).

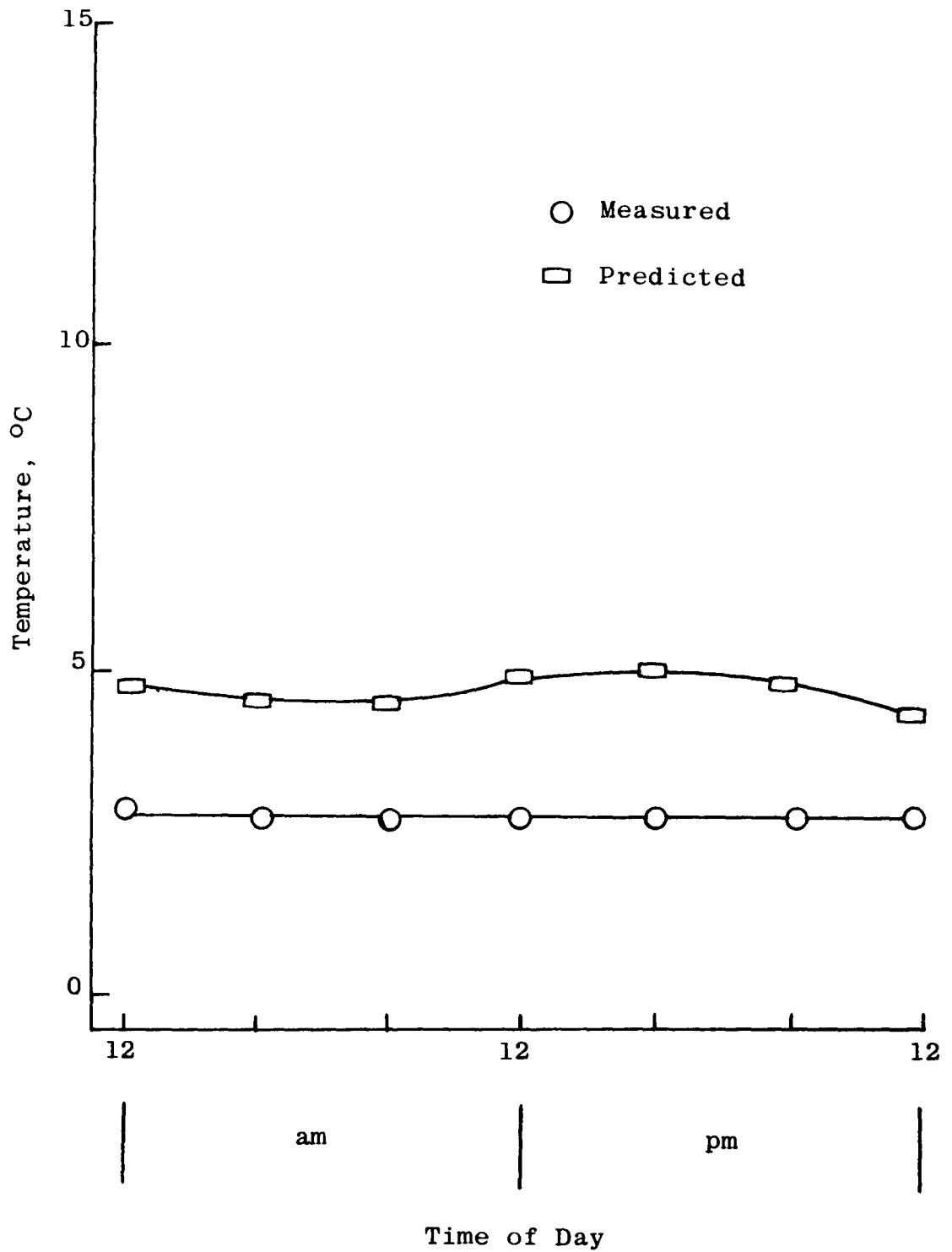


Figure 18. Comparison of predicted and measured waste treatment lagoon temperatures (December 1).

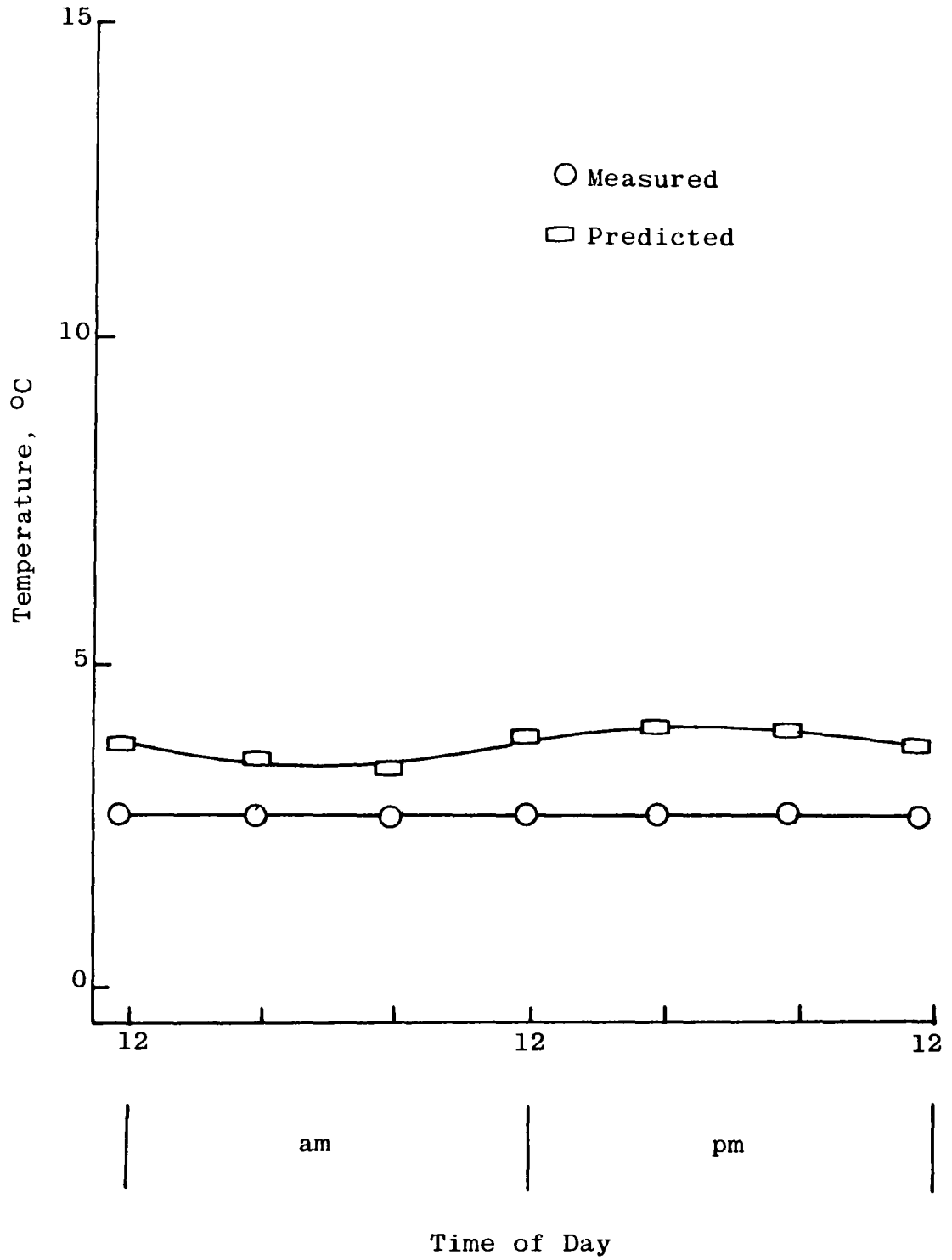


Figure 19. Comparison of predicted and measured waste treatment lagoon temperatures (December 12).

TABLE VIII. Comparison to energy loss to temperature change in the waste treatment lagoon as predicted by the computer program.

<u>Date</u>	<u>Time of Day</u>	<u>Energy Loss (<math>10^7</math> J)*</u>	<u>Temperature Change, °C</u>
Oct 12	6 - 7 (am)	36.65	-0.06
	8 - 9	-22.06 <sup>1</sup>	+0.03
	10 - 11	-64.31	+0.11
	12 - 1	-76.31	+0.13
	2 - 3	-54.90	+0.09
	4 - 5	- 6.44	+0.01
	6 - 7	42.84	-0.07
Nov 3	6 - 7 (am)	35.96	-0.06
	8 - 9	-18.47	+0.03
	10 - 11	-64.66	+0.11
	12 - 1	-78.18	+0.13
	2 - 3	-55.04	+0.09
	4 - 5	- 2.81	+0.01
	6 - 7	40.10	+0.07
Dec 1	6 - 7	31.64	-0.05
	8 - 9	-14.04	+0.02
	10 - 11	-65.27	+0.11
	12 - 1	-80.62	+0.14
	2 - 3	-54.83	+0.09
	4 - 5	2.53	-0.01
	6 - 7	35.46	-0.06

\*To convert from J to BTU, divide by 1055

<sup>1</sup>Negative value means energy is being supplied to the lagoon

## SUMMARY AND CONCLUSIONS

A project located at the VPI & SU Swine Center was used to study the feasibility of utilizing a waste treatment lagoon as an energy source for heating livestock shelters. To aid in this investigation, a computer model to simulate lagoon response was developed. The model predicted energy gains due to solar insolation and heat losses due to convection, conduction, evaporation and radiation. Values obtained in the simulation were used to estimate the available energy in the lagoon and the effects of energy removal on lagoon stability.

Conclusions drawn from the computer simulation and performance testing of the system were:

1. Low level energy is available in the waste treatment lagoon. A small change in the overall lagoon temperature requires a significant amount of energy (Table VIII). For example, a temperature decrease of  $0.06^{\circ}\text{C}$  ( $0.11^{\circ}\text{F}$ ) on October 12 between 6-7 a.m. required  $3.7 \times 10^8\text{ J}$  ( $3.5 \times 10^5\text{ BTU}$ ) of energy removal. This amount of energy is equivalent to 102.5 kwh of electric energy or 9.8 liters (2.6 gallons) of kerosene.
2. The low level energy available in the waste treatment lagoon may be removed to heat swine housing.

As shown in Figure 16, energy removal was possible when the working fluid temperature was greater than  $4.4^{\circ}\text{C}$  ( $40^{\circ}\text{F}$ ). After energy was removed to heat the nursery on October 26, the working fluid was circulated through the heat exchanger to gain energy from the lagoon. After a short period of time, the working fluid returned to equilibrium with the lagoon temperature. Using this process, the lagoon supplied energy in sufficient quantities to maintain the nursery temperature at  $29.4^{\circ}\text{C}$  ( $85^{\circ}\text{F}$ ). If the working fluid temperature dropped below  $4.4^{\circ}\text{C}$  ( $40^{\circ}\text{F}$ ), energy could no longer be supplied by the lagoon to maintain the nursery temperature at  $29.4^{\circ}\text{C}$  ( $85^{\circ}\text{F}$ ). Therefore, modifications to the heat pump are required to allow operation utilizing a working fluid with a temperature below  $4.4^{\circ}\text{C}$  ( $40^{\circ}\text{F}$ ).

3. The removal of energy from the waste treatment lagoon has little effect upon lagoon thermal stability. As shown in Table VIII, the corresponding energy loss versus temperature change was high. For thermal instability to occur, a temperature change greater than those calculated with the computer simulation and those recorded during system testing must occur.

## RECOMMENDATIONS

The following recommendations are made concerning the computer simulation and the utilization of the waste treatment lagoon to heat swine housing.

1. To more accurately predict lagoon response, the computer model should be refined to include modifications for variations in cloud cover, wind speed, dust conditions and daily temperature. A subroutine model of the psychrometric chart for use in calculating evaporative loss is also needed.
2. The feasibility of reducing losses from the lagoon due to convection, radiation and evaporation should be evaluated. Investigations should include (a) tests to reduce energy loss by decreasing the surface area of the lagoon exposed to the atmosphere, (b) tests of insulating the entire lagoon to reduce energy loss and (c) evaluations of surface coverings to enhance solar insolation absorption.
3. The water-to-air heat pumps now available commercially should be modified to handle a working fluid with a temperature less than  $4.4^{\circ}\text{C}$  ( $40^{\circ}\text{F}$ ). Experimentation with the proper modifications would allow for the development of a system which could utilize the low level energy available in a working

fluid with a temperature of 4.4° C (40° F) or less without damage to the heat pump. This area of work appears to be the limiting factor in utilizing a waste treatment lagoon as an energy source for heating swine housing.

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APPENDIX A

COMPUTER PROGRAM

(A complete copy of the program and generated data is available for inspection at the VPI & SU Agricultural Engineering Department.)

C  
C THIS SECTION CALCULATES THE AVAILABLE SOLAR ENERGY FOR THE YEAR  
C AND BREAKS THE CALCULATIONS INTO VALUES FOR EACH DAY , HOUR , AND  
C MINUTES FOR THIS PERIOD. UNITS ARE BTU / SQ.FT. DAY.  
C  
C DAWN -- SUNRISE , MINUTES PAST MIDNIGHT  
C DUSK -- SUNSET , MINUTES PAST MIDNIGHT  
C QHD -- HOURLY INSOLATION RATE  
C LT -- LAGOON TEMP.  
C TT -- TANK TEMP.  
C TA -- AIR TEMP.  
C TG -- GROUND TEMP.  
C CF -- CLOUD FACTOR  
C L -- LENGTH OF LAGOON  
C W -- WIDTH OF LAGOON  
C D -- DEPTH OF LAGOON  
C FLOW -- FLOW OF WORKING MEDIUM , GPM  
C RFLOW -- FLOW OF WORKING MEDIUM , FT./SEC.  
C FLOWP -- FLOW OF WORKING MEDIUM , LB./MIN.  
C A -- LAGOON AREA  
C VOL -- LAGOON VOLUME  
C DIAIN -- INSIDE DIAMETER OF TUBING , FT.  
C DIAOT -- OUTSIDE DIAMETER OF TUBING , FT.  
C DEN -- DENSITY OF WATER , LB./CU.FT.  
C EXL -- LENGTH OF EXCHANGER , FT.  
C FCOU -- OUTSIDE FILM COEFFICIENT  
C FCIN -- INSIDE FILM COEFFICIENT  
C R -- THERMAL CONDUCTIVITY OF PLASTIC TUBING  
C LHS -- LATENT HEAT OF STEAM  
C DP -- DIFFERENCE IN SAT. VAPOR PRESSURE AND ACTUAL VAPOR PRESSURE  
C V -- WIND VELOCITY , MPH  
C VA -- AIR VELOCITY , FT./SEC.

C DA -- DENSITY OF AIR , 0.075 LB./CU.FT.  
C HC -- THERMAL CONDUCTIVITY OF SOIL , 0.025 - 0.035  
C HCV -- CONVECTIVE TRANSFER COFF.  
C CP -- 1.0 BTU/LBM./DEG.F.  
C DT -- TEMP. DIFFERENCE  
C DT1 -- TEMP. DIFFERENCE  
C SOL -- AVAILABLE HOURLY SOLAR INSOLATION  
C ADDHL -- ADDED HEAT TO THE LAGOON  
C LTH -- HOURLY LAGOON TEMP.  
C TTH -- HOURLY TANK TEMP.  
C QCOND -- CONDUCTIVE LOSSES  
C QCONV -- CONVECTIVE LOSSES  
C QR -- RADIATION LOSSES  
C QE -- EVAPORATIVE LOSSES  
C QTN -- NATURAL LOSSES  
C QEX -- EXCHANGER LOSSES  
C TADDHL -- TOTAL ADDED HEAT TO THE LAGOON  
C TQCOND -- TOTAL CONDUCTIVE LOSS  
C TQCONV -- TOTAL CONVECTIVE LOSS  
C TQR -- TOTAL RADIATIVE LOSS  
C TQE -- TOTAL EVAPORATIVE LOSS  
C TQTN -- TOTAL NATURAL LOSS  
C TQEX -- TOTAL EXCHANGER LOSS  
C TSUML -- TOTAL LOSSES  
C KVIS -- KINEMATIC VISCOSITY  
C THCON -- THERMAL CONDUCTIVITY  
C REYNO -- REYNOLDS NO.  
C PRNO -- PRANDTL NO.  
C DNUNO -- NUSSELT'S NO.  
C U -- HEAT TRANSFER COFF. OF PLASTIC TUBING  
C TLEX -- TEMP. OF WORKING MEDIUM LEAVING THE EXCHANGER  
C

```

REAL MARC
REAL LT,LTH,L,LHS,KVIS
DIMENSION TA(365)
DIMENSION QHD(365,24),DAWN(365),DUSK(365)
DIMENSION TADDHL(365,24),TQCOND(365,24),TQCONV(365,24)
DIMENSION TQR(365,24),TQE(365,24),TQTN(365,24),TQEX(365,24)
DIMENSION LTH(365,24),TSUML(365,24),TTH(365,24)
DO 11 I=1,365
  IF(I.LE.86) GO TO 50
  IF(I.GT.86.AND.I.LE.280) GO TO 60
  IF(I.GT.280) GO TO 55
50 DAWN(I)=375.0-((76.0/86.0)*I)
   DUSK(I)=397.0+((66.0/86.0)*I)+720.
   GO TO 71
60 DAWN(I)=230.0+((155.0/194.0)*I)
   DUSK(I)=523.0-((150.0/194.0)*I)+720.
   GO TO 71
55 DAWN(I)=715.0-((79.0/85.0)*I)
   DUSK(I)=32.0+((84.0/85.0)*I)+720.
   GO TO 71
71 IF(I.LE.274) GO TO 70
   IF(I.GT.274) GO TO 75

```

```

C
C Q EQUALS THE TOTAL DAILY RADIATION FOR A CLEAR DAY . THE UNITS ARE
C W/SQ. M. DAY. QD IS THE CONVERSION OF Q TO BTU / SQ. FT. DAY.
C I EQUALS THE NO. OF DAYS. J EQUALS THE NO. OF HOURS. K EQUALS THE
C NO. OF MINUTES.

```

```

C
70 Q=9000.00*SIN((3.14*I/365.0)+(3.14/4.0))+21000.0
   GO TO 81
75 Q=9000.00*SIN((3.14*I/365.0)+(5.*3.14/4.))+21000.00
   GO TO 81

```

```

      81 QD=Q*((1000.00*(.3048**2.0))/1055.8)
        DD=DUSK(I)-DAWN(I)
        DUM=DAWN(I)
        IDUM=IFIX(DAWN(I))
        XDUM=IDUM
        DUM=DAWN(I)-XDUM
C
C FIND CONSTANT
C INTEGRATE FROM 0 TO DD
C
      AREA=0.0
      IN=1000
      DO 40 JJ=1,IN
        W=DD/IN
        WHALF=W/2.
        H=((JJ*W)-WHALF)*3.14159/DD)
        AREA=(W*H)+AREA
40 CONTINUE
      CONST=2.45*QD/AREA
      SUM=0
      DO 10 J=1,24
        QHD(I,J)=0.0
      DO 12 K=1,60
        M=(J-1)*60+K
        IF(M.LT.DAWN(I)) GO TO 12
        IF(M.GT.DUSK(I)) GO TO 12
        XM=M+DUM-0.5-DAWN(I)
C
C DAY EQUALS THE HOURLY CALCULATION OF SOLAR RADIATION.( BTU / SQ. FT. )
C
      DAY=CONST*SIN((XM*3.14159)/DD)
      QHD(I,J)=QHD(I,J)+DAY

```

```

12 CONTINUE
  SUM=SUM+QHD(I,J)
10 CONTINUE
11 CONTINUE
  DO 3000 I IIII=1,365
    TA(IIII)=20.2*SIN(3.14*IIII/182.5)+50.0
3000 CONTINUE

```

```

C
C THIS SECTION OF THE PROGRAM CALCULATES THE HEAT LOSSES OR GAINS
C FROM THE LAGOON BY NATURAL OR ARTIFICIAL MEANS .
C

```

```

  READ(5,700)LT,CF,L,W,D,TG
700 FORMAT(6F6.2)
  READ(5,701)TT,FLOW,DIAIN,DI AOT,CP,DEN,EXL
701 FORMAT(7F6.2)
  READ(5,702)FCOUT,R
702 FORMAT(2F10.6)
  READ(5,703)LHS,DP,V,DA,HC
703 FORMAT(5F10.6)
  READ(5,704)X
704 FORMAT(F6.2)
  A=L*W
  VOL=A*D
  DIAIN=DI AIN/12.0
  DIAOT=DI AOT/12.0
  RFLOW=8.34*FLOW/(DEN*60.0)
  FLOWP=FLOW*8.34
  VA=1.466*V
  E=0.95
  WRITE(6,496)
496 FORMAT(////,' DATA ',/)
  WRITE(6,800)

```

```

800 FORMAT(/,7X,'LT          CF          L          W          D          TG
9  ')
WRITE(6,499)LT,CF,L,W,D,TG
499 FORMAT(6F10.2)
WRITE(6,801)
801 FORMAT(/,6X,'TT          FLOW          DIAIN          DIAOT          CP          DEN
8  EXL')
WRITE(6,498)TT,FLOW,DIAIN,DIAT,CP,DEN,EXL
498 FORMAT(7F10.4)
WRITE(6,802)
802 FORMAT(/,3X,' FCOUT          R          LHS          DP          V
7DA          HC          X')
WRITE(6,497)FCOUT,R,LHS,DP,V,DA,HC,X
497 FORMAT(8F10.4)
DO 200 N=190,365
DO 201 II=1,24
TADDHL(N,II)=0
TQCOND(N,II)=0.0
TQCONV(N,II)=0.0
TQR(N,II)=0.0
TQE(N,II)=0.0
TQTN(N,II)=0.0
TQEX(N,II)=0.0
LTH(N,II)=0.0
TSUML(N,II)=0.0
TTH(N,II)=0.0
DO 202 KK=1,60
MM=(II-1)*60+KK
IF(MM.LT.DAWN(N).OR.MM.GE.DUSK(N)) GO TO 300

```

```

C
C ADDITIONS FROM SOLAR INSOLATION
C

```

```

      SOL=CF*A*(QHD(N, II)/60.0)
      ADDHL=0.9*SOL
      GO TO 301
C
C  LOSSES BY NATURAL MEANS
C
C  CONDUCTIVE LOSS TO GROUND
C
300  ADDHL=0.00
301  DT=LT-TG
      QCOND=(HC*A*DT)/60.0
C
C  CONVECTIVE LOSS TO AIR
C
      DT1=LT-TA(N)
      DT2=ABS(DT1)
      IF(V.LE.1.00) HCV=0.38*(DT2**0.25)
      IF(V.GT.1.00.OR.V.LE.3.00) HCV=1.00
      IF(V.GT.3.00.OR.V.LE.5.00) HCV=2.00
      IF(V.GT.5.00.OR.V.LE.7.50) HCV=3.00
      IF(V.GT.7.50) HCV=4.00
      QCONV=(HCV*A*DT1)/60.0
C
C  RADIATION LOSSES
C
      QR={(0.173*A*E*(((LT+360.0)/100.0)**4.0)-(((TA(N)+360.0)/100.0)**4.0)))/60.0
C
C  EVAPORATION LOSSES
C
      QE=(1.08*((VA*DA)**0.8)*DP*LHS*A)/60.0
C

```

C TOTAL LOSS BY NATURAL MEANS

C

QTN=QCOND+QCONV+QR+QE-ADDHL  
LT=LT-(QTN/(VOL\*DEN))

C

C EXCHANGER LOSSES

C

KVIS=0.000456/(TT\*\*0.90458)  
THCON=2.8175\*(TT\*\*0.09551)  
REYNO=(2.0\*RFLOW)/(3.14\*(DIAIN/2.0)+KVIS)  
PRNO=(KVIS\*DEN\*CP)/THCON  
DNUNO=0.023\*(REYNO\*\*0.8)\*(PRNO\*\*0.035)  
FCIN=(DNUNO\*THCON)/(2.0\*(DIAIN/2.0))  
U=1.0/((1.0/FCIN)+(1.0/FCOUT)+(1.0/R))  
TLEX=((LT-TT)\*(1.0/EXP((6.28\*(DIAOT/2.0)\*EXL\*U)/(FLOWP\*CP))))+TT  
QEX=FLOWP\*CP\*(TLEX-TT)  
TT=TT+(QEX/500.4)  
LT=LT-(QEX/(VOL\*DEN))  
LTH(N,II)=LT  
TTH(N,II)=TT  
TADDHL(N,II)=TADDHL(N,II)+ADDHL  
TQCOND(N,II)=TQCOND(N,II)+QCOND  
TQCONV(N,II)=TQCONV(N,II)+QCONV  
TQR(N,II)=TQR(N,II)+QR  
TQE(N,II)=TQE(N,II)+QE  
TQTN(N,II)=TQTN(N,II)+QTN  
TQEX(N,II)=TQEX(N,II)+QEX  
TSUML(N,II)=TQEX(N,II)+TQTN(N,II)

202 CONTINUE

201 CONTINUE

200 CONTINUE

WRITE(6,502)

```
502 FORMAT(///,2X,' ADD HEAT ',2X,' CONDUCTIVE',3X,' CONVECTIVE',4X,' RA  
2DIATION',1X,' EVAPORATION',6X,' EXCHANGER',7X,' TOTAL LOSS',3X,' LAG  
3TEMP ',3X,' TANK TEMP ',1X,' DAY',/)  
DC 1000 NN=190,365  
DG 2000 NNN=1,24  
WRITE(6,503)TADDHL(NN,NNN),TQCOND(NN,NNN),TQCONV(NN,NNN),TOR(NN,NN  
1N),TQE(NN,NNN),TQEX(NN,NNN),TSUML(NN,NNN),LTH(NN,NNN),TTH(NN,NNN),  
2NN,NNN  
503 FORMAT(1X,9F12.2,1X,I3,1X,I3)  
2000 CONTINUE  
1000 CONTINUE  
STOP  
END
```

## Appendix B

### Corresponding Computer Values for Days in the Year

## Corresponding Computer Values for Days in the Year

Date	Month				
	January	February	March	April	May
1	282	313	341	7	37
2	283	314	342	8	38
3	284	315	343	9	39
4	285	316	344	10	40
5	286	317	345	11	41
6	287	318	346	12	42
7	288	319	347	13	43
8	289	320	348	14	44
9	290	321	349	15	45
10	291	322	350	16	46
11	292	323	351	17	47
12	293	324	352	18	48
13	294	325	353	19	49
14	295	326	354	20	50
15	296	327	355	21	51
16	297	328	356	22	52
17	298	329	357	23	53
18	299	330	358	24	54
19	300	331	359	25	55
20	301	332	360	26	56
21	302	333	361	27	57
22	303	334	362	28	58
23	304	335	363	29	59
24	305	336	364	30	60
25	306	337	365	31	61
26	307	338	1	32	62
27	308	339	2	33	63
28	309	340	3	34	64
29	310	---	4	35	65
30	311	---	5	36	66
31	312	---	6	--	67

## Corresponding Computer Values for Days in the Year

Date	Month				
	June	July	August	September	October
1	68	98	129	160	190
2	69	99	130	161	191
3	70	100	131	162	192
4	71	101	132	163	193
5	72	102	133	164	194
6	73	103	134	165	195
7	74	104	135	166	196
8	75	105	136	167	197
9	76	106	137	168	198
10	77	107	138	169	199
11	78	108	139	170	200
12	79	109	140	171	201
13	80	110	141	172	202
14	81	111	142	173	203
15	82	112	143	174	204
16	83	113	144	175	205
17	84	114	145	176	206
18	85	115	146	177	207
19	86	116	147	178	208
20	87	117	148	179	209
21	88	118	149	180	210
22	89	119	150	181	211
23	90	120	151	182	212
24	91	121	152	183	213
25	92	122	153	184	214
26	93	123	154	185	215
27	94	124	155	186	216
28	95	125	156	187	217
29	96	126	157	188	218
30	97	127	158	189	219
31	--	128	159	---	220

## Corresponding Computer Values for Days in the Year

Date	Month	
	November	December
1	221	251
2	222	252
3	223	253
4	224	254
5	225	255
6	226	256
7	227	257
8	228	258
9	229	259
10	230	260
11	231	261
12	232	262
13	233	263
14	234	264
15	235	265
16	236	266
17	237	267
18	238	268
19	239	269
20	240	270
21	241	271
22	242	272
23	243	273
24	244	274
25	245	275
26	246	276
27	247	277
28	248	278
29	249	279
30	250	280
31	---	281

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UTILIZATION OF A WASTE TREATMENT  
LAGOON AS AN ENERGY SOURCE

by

Carlton Lee Hill

(ABSTRACT)

A project located at the VPI & SU Swine Center was used to investigate the utilization of a waste treatment lagoon as an energy source to heat swine housing. The investigation test procedure consisted of two main components: (1) the design and testing of a heating system using a solar-assisted heat pump and (2) the development of a computer model to simulate lagoon response to energy gains and losses. The energy lost by the lagoon was considered to occur by conduction, convection, evaporation and radiation. Under natural weather conditions, the losses due to these factors become gains when the lagoon temperature is less than the temperature of the surrounding earth and air. Energy gain by the lagoon under most operating conditions was due to solar insolation.

By using the information generated by the computer model and data collected during the testing of the system, it was concluded that there is low level energy available in the waste treatment lagoon. Also, the removal of the low level energy for heating swine housing has little effect upon the thermal stability of the lagoon.