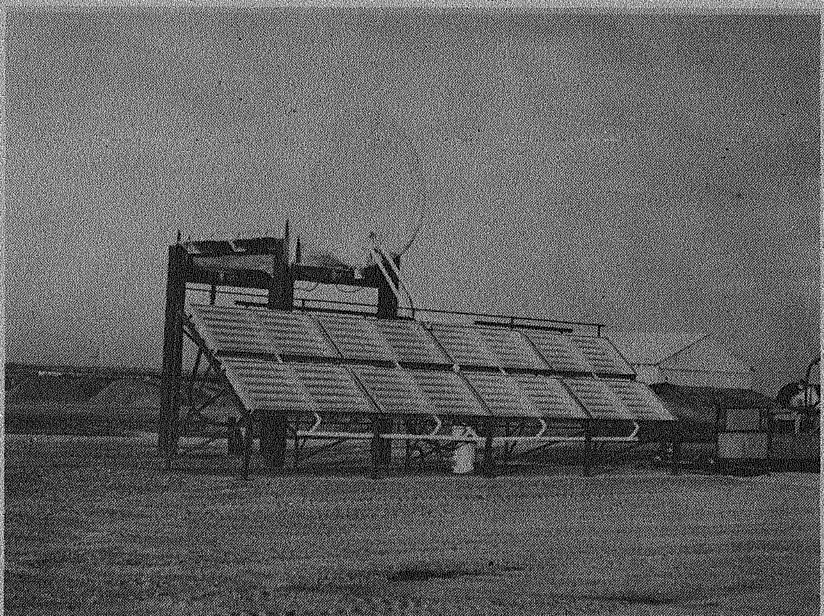
SOLAR ENERGY
HEATING OF ASPHALT to 200°F
in storage tanks

Final Report
prepared for
STATE OF OKLAHOMA
DEPARTMENT OF TRANSPORTATION



SCHOOL OF MECHANICAL AND
AEROSPACE ENGINEERING
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

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16. Abstract

A high-temperature (185 - 225°F) asphalt storage system has been designed and constructed at Perry, Oklahoma. This report documents the design process, construction procedures, and evaluates data gathered to date. The design is covered in detail including sizing the auxiliary heater, estimating heat losses, sizing collectors, and designing controls. The construction stage is documented including air bleed system problems, collector protection, and start-up procedures. Data gathered for a fifteen day period in December, 1979 is compared directly to the computer model used to design the solar system. Asphalt temperatures are also recorded for the period from Dec. 1, 1979 to Jan. 15, 1980.

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SOLAR ENERGY HEATING
OF ASPHALT TO 200°F
IN STORAGE TANKS

FINAL REPORT

Submitted to

RESEARCH DIVISION

STATE OF OKLAHOMA

DEPARTMENT OF TRANSPORTATION

OKLAHOMA CITY

by

School of Mechanical & Aerospace Engineering
Oklahoma State University
Stillwater

Dr. Jerald D. Parker
Dr. John A. Wiebelt
Mr. Gary B. Ferrell

June 18, 1980

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CHAPTER I

INTRODUCTION

Asphalt used for road repairs by Oklahoma Department of Transportation personnel must be stored within prescribed temperature limits. Asphalt emulsions must be stored at 65-160°F. Cutback asphalts must be stored at 185-225°F. Asphalt emulsions will break apart if either temperature limit is exceeded. Cutback asphalts will either harden excessively or separate.

Solar heated storage systems designed to store the asphalt emulsions (65-160°F) have been quite successful to date. The ODOT Muskogee facility has demonstrated the relatively economical performance of these low temperature systems. Because of very low flat plate collector efficiencies at temperatures approaching 200°F, few attempts have been made to use solar energy for heating cutback asphalt tanks.

The objective of this report is to examine the design, construction, and performance of a solar system designed to store the higher temperature cutback asphalts. This particular system is now in operation at the ODOT division headquarters in Perry, Oklahoma. Figure 1 offers two views of the storage tank and collector array.

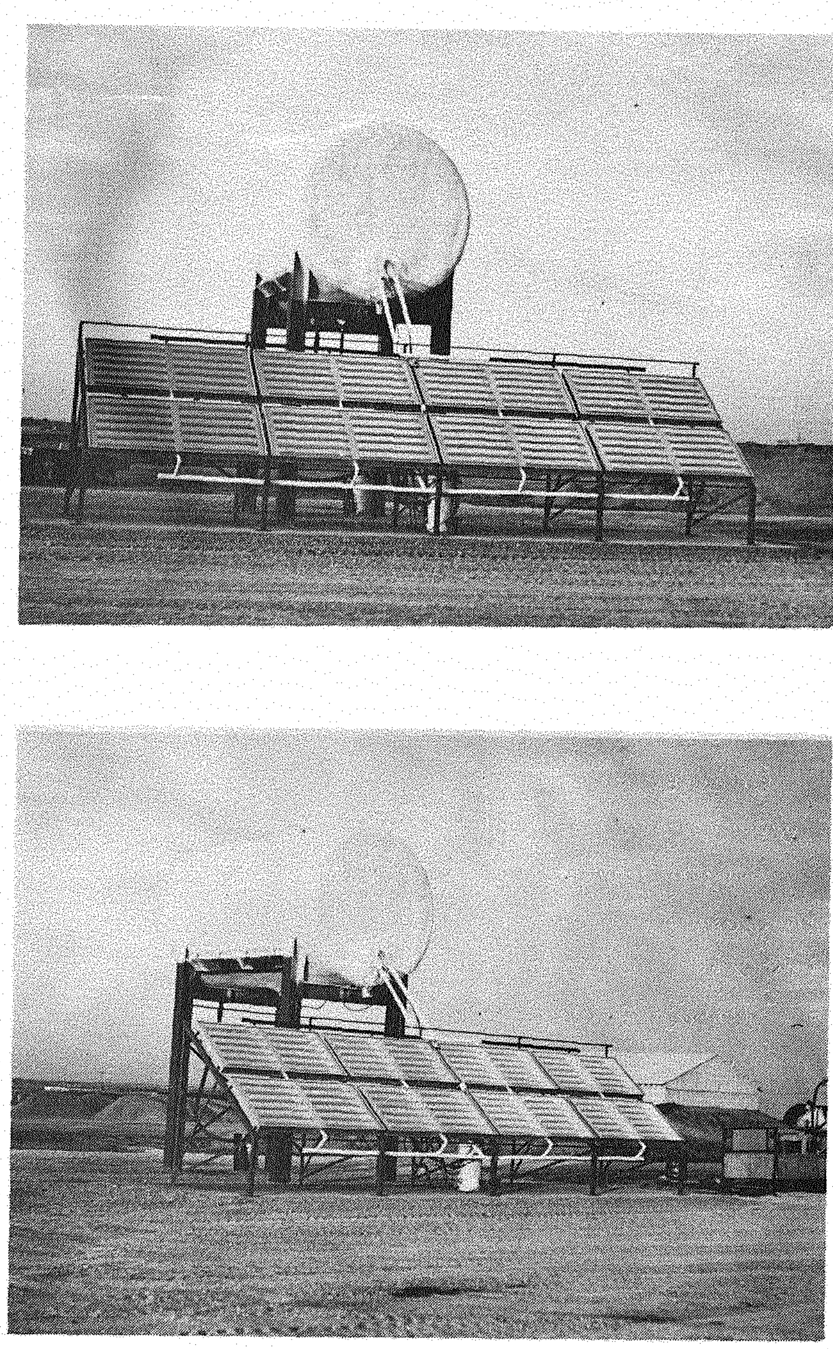


FIGURE 1

The solar heated asphalt storage system located at the Oklahoma Department of Transportation Division Four maintenance yard in Perry, Oklahoma. Photographs taken in late December, 1979.

CHAPTER II

DESIGN

Economics usually is a major factor in the design of a high-temperature asphalt storage system. The large capital outlay for solar collectors and controls require that a design be optimum, based upon both experience and analysis of thermal performance.

The solar heated storage tank at Perry was meant primarily for research purposes. No attempt was made to optimize the system economically. The system was designed for practical use and long life. The experiences gained at Perry should prove quite beneficial when constructing similar systems.

SYSTEM CONCEPT

It was desired to maintain 10,000 gallons of cutback asphalt at 185 to 225°F. An auxiliary heater was provided to maintain temperatures when solar energy gathered by the collectors was unable to meet the load. It was important to simplify the system as much as possible without endangering components or hampering performance. The flow schematic in figure 2 illustrates how the system at Perry functions. In addition to meeting the thermal loads, the flow system and related controls must also protect the collectors. Without an energy dissipation system, the collector area should not be so large as to overheat the asphalt.

HEAT LOSSES

To size the auxiliary heater properly, the worst-case system heat losses were estimated. This estimation was later used when sizing the collectors.

Using experience gained in the design of a solar system at Clinton, Oklahoma (1), it was decided to insulate the Perry asphalt tank with 5 inches of polyurethane foam. The worst case design condition for Perry was considered to be a 0°F night while trying to maintain 200°F asphalt temperatures.

HEAT LOSSES FROM TANK

$$\dot{Q}_{tank} = \Delta T \frac{A}{R}$$

ΔT = temperature difference = 200°F

A = surface area of tank and insulation at half thickness

$$A = \frac{1}{2}\pi D^2 + \pi DL$$

D = diameter of tank plus half thickness of insulation

D = 10.417 ft.

L = length of tank plus half thickness of insulation

L = 19.417 ft.

A = 806 ft^2

R = thermal resistance of tank insulation

$$R = \frac{t}{k}$$

t = thickness of insulation = 5 inches

k = thermal conductivity of insulation

For this variable, refer to appendix III

k = 0.020 BTU/hr ft °F

R = 20.83 hr-ft^2/BTU

Therefore,

$$\dot{Q}_{tank} = 7738 \text{ BTU/hr}$$

Note that the above value assumed that there were no penetrations of the insulation. The Perry system was designed so that the tank would have no appreciable penetrations to increase heat loss. The tank was isolated from its support cradle by thick rubber pads. The support framework adjacent to the tank was sprayed with foam when the tank was insulated. The manway at the top was insulated also. If it should ever be necessary to enter the tank, the insulation can be cut away and then repaired after entry.

If a manway had been left uninsulated, the following calculation would have been used. The top manway would be considered as a flat plate with convective heat transfer to the surrounding air. The equation for calculating the heat transfer coefficient (2) to the surrounding air was:

$$h = 0.38\ (\Delta T)^{1/4}\ \text{BTU/hr ft}^2\ °\text{F}$$

$$\Delta T = 200°\text{F} = 200°\text{R}$$

$$h = 1.43\ \text{BTU/hr ft}^2\ °\text{F}$$

For an 18 inch diameter manway mounted on a 200°F tank (assume steel is at 200°F):

$$\dot{Q}_{manway} = hA\ \Delta T$$

$$A = 1.77\ \text{ft}^2$$

$$\Delta T = 200°\text{F}$$

$$\dot{Q}_{manway} = 505.4\ \text{BTU/hr}$$

Note that if the manway were left uncovered, this would be a sizeable loss.

HEAT LOSSES FROM FLUID PIPING

At Perry, worst case for piping heat loss was determined to be losses from approximately 130 ft of one inch copper tubing through 3/4 inch thick rubber foam such as Armaflex. Average fluid temperature was assumed to be 200°F. From appendix III, it can be seen that the approximate thermal conductivity of rubber foam is 0.016 BTU/hr ft °F.

$$\dot{Q}_{pipe} = \Delta T\ \frac{A}{R}p$$

$$\Delta T = 200°\text{F}$$

$$A_p = L_p \pi\ d_o$$

$L_p = 130$ ft

$d_o = 2.5/12$ ft

$A_p = 85.1\ ft^2$

$R = R_i + R_o$

$$R_i = \frac{d_o \ln {d_o/d_i}}{2k}$$

$d_o = 2.5/12$ ft

$d_i = 1.0/12$ ft

$k = 0.016$ BTU/hr ft °F

$R_i = 5.97$ hr °F/BTU

$R_o = R_{film} \simeq 1/3$ (average wind conditions)

$R \simeq 6.3$ hr °F/BTU

$\dot{Q}_{pipe} = 2701$ BTU/hr

TOTAL HEAT LOSS (worst condition)

$\dot{Q}_{total} = \dot{Q}_{tank} + \dot{Q}_{pipe}$

$\dot{Q}_{total} = 10439$ BTU/hr = 3.06 KW

In this manner it was determined that the minimum size auxiliary heater for Perry should be 4 KW. This heater, installed in the expansion tank, was deemed adequate to maintain 200°F asphalt temperature on a 0°F ambient temperature night.

The collectors were sized using the heat losses along with several parameters. The procedure was like the above for sizing the heaters except that the temperature difference (ΔT) was varied in accordance with weather conditions on an hour-by-hour basis. The iterative computer simulation used may be found in appendix I.

COLLECTOR SELECTION

One of the reasons for the construction of the system at Perry was to evaluate the use of evacuated glass tube concentrating collectors.

Concentrating collectors focus the sun's rays to a receiving tube. Because this area is smaller than a flat-plate collector of the same total aperture, conduction and radiation losses are minimized. This boosts operating temperatures. Placing the receiver inside of an evacuated tube practically eliminates conduction losses. All of the evacuated tube collectors considered were stationary.

Three collector makes were considered. They were the Sunmaster by Owens-Illinois, the TC-100 by General Electric, and the XE-300 by Energy Design Corporation. Because of thermal stress and possible outgassing problems all three manufacturers recommended temperature protection. The many parameters forcing the final purchase included efficiency over the temperature range, durability, ability to drain down, and cost. The Energy

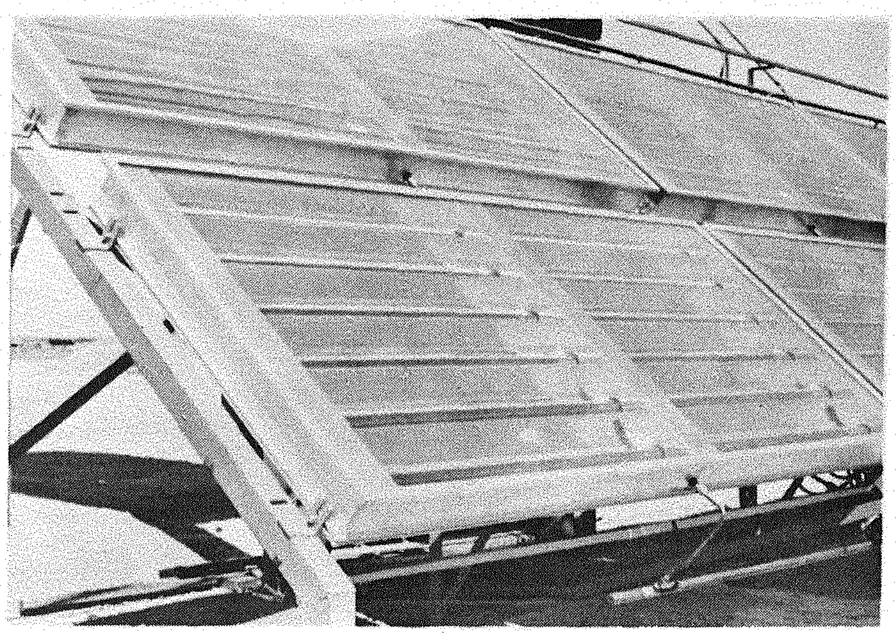


FIGURE 3

The Energy Design XE-300 collectors as installed at Perry. Note the piping connections.

Design XE-300 proved the most versatile for this application, combining high-temperature efficiency and rugged design.

The XE-300 is shown in a photo in figure 3. Note that the evacuated tubes run horizontally and have aluminum reflectors behind them. The glass cover over tubes and reflectors was seen to be advantageous for dusty maintenance yard conditions. The efficiency curves provided by the manufacturer are shown in figure 4. The collector efficiency equation is included in the computer program in appendix I.

COLLECTOR SIZING AND ORIENTATION

Determination of the area of collection depends upon several factors. At Perry, it was desired to size the collector area such that solar energy carried about half the load on an annual basis. Early in the design process it was estimated that an array of approximately 250 ft^2 would accomplish this. This estimation followed from simulations on Oklahoma State University's IBM 370 computer. Later, a complete computer simulation evolved as the system took shape. This simulation utilized hourly weather data and load calculations to evaluate different parameters. Again, this program may be found in appendix I. The final area decided upon was an even number of Energy Design XE-300 collectors -- eight. At 28 ft^2 of aperture area per collector, the total collection area was set at 224 ft^2.

Collector orientation was optimized to deliver the most energy to the asphalt tank on an annual basis. The reflectors on the collectors have an acceptance angle of 72°. If a normal is projected from the collector face, the sun's rays projected in a north-south vertical plane must be within 36° of this normal -- to the north or to the south. Incorporating this into computer simulations, a collector tilt angle of 40° from horizontal was determined to be optimum. Note that for year-round heating with flat-plate collectors, the tilt angle is approximately the latitude (Perry -- 36.3°)(3).

INSTANTANEOUS EFFICIENCY OF THE ENERGY DESIGN CORPORATION
XE-300 SOLAR ENERGY COLLECTOR

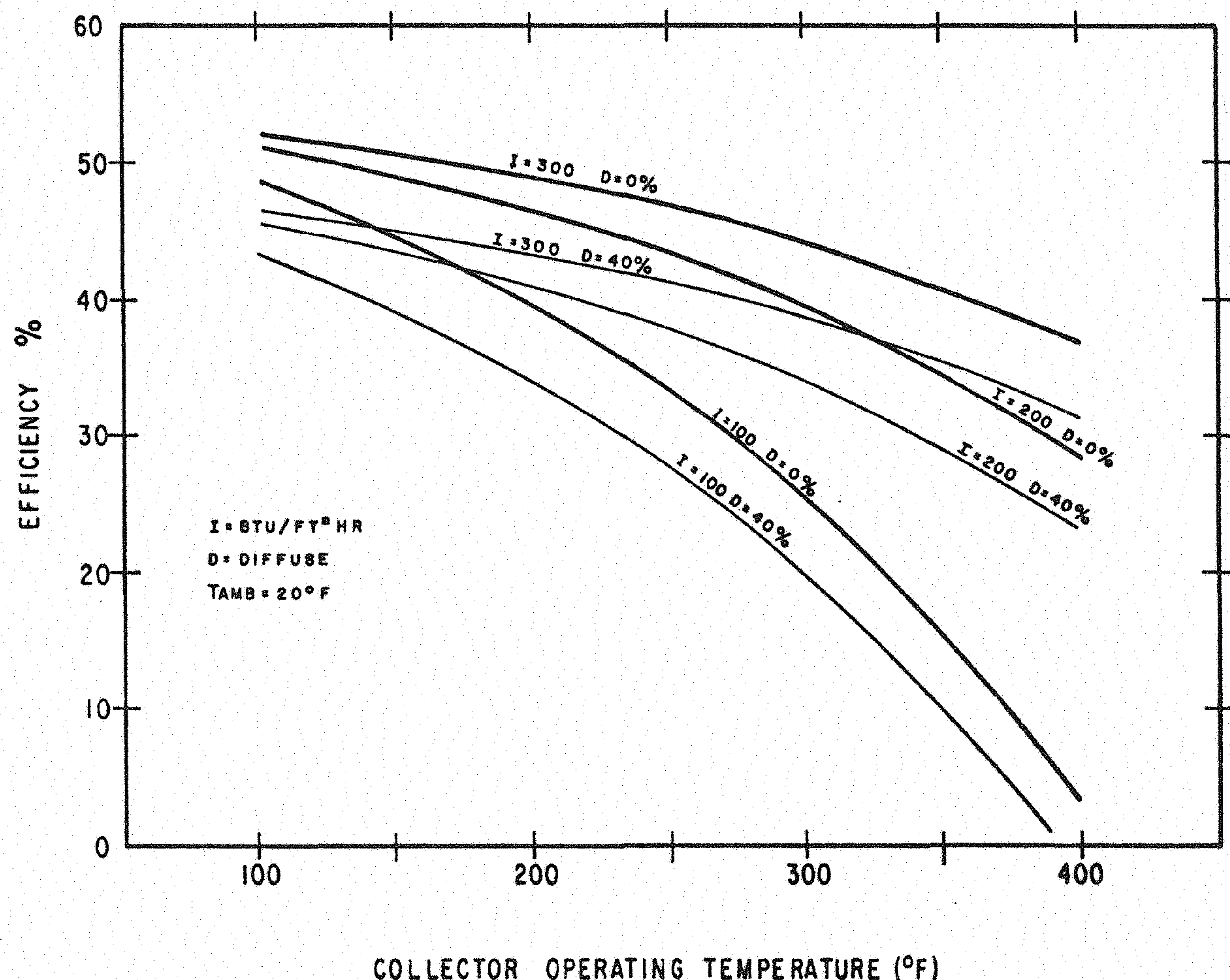


FIGURE 4

The most efficient collector arrangement is to place them all in parallel. This scheme keeps the average collector temperature as low as possible, the most efficient condition. For the Perry system, it was decided to arrange the eight collectors in four parallel series pairs, as can be seen in Figure 1. This arrangement takes a lot less space and piping than all eight in parallel. Heat loss from the extra pipe in an all-parallel arrangement may cancel any efficiency advantages.

In addition to properly orienting the collectors, the collector support stands must also be structurally sound and simple to construct. At Perry, the stands were designed to raise the bottom of the lowest collector above the maximum fluid level in the expansion tank. This permits complete draining of the system. Wind speeds occasionally exceed 75 mph at the Perry site, and the stands were designed accordingly. Figures 5 and 6 illustrate the stands and site arrangement.

HEAT EXCHANGER DESIGN

The overall performance of the solar system depends heavily upon the effectiveness of the heat exchanger in the asphalt tank. The effectiveness should approach 1.0 as closely as economically feasible. Allowing heat exchanger exit temperatures to greatly exceed the bulk temperature of the asphalt drastically cuts thermal performance of the entire system.

At Perry, it was decided to use a finned tube heat exchanger. Eight passes sixteen feet long were chosen. The tubes are constructed of admiralty brass one inch in diameter. Aluminum fins 5/8 inch in height are mechanically wrapped ten per inch. Figure 7 illustrates how the heat exchanger was designed to rest in the bottom of the tank. This was done to prevent cold asphalt from settling below the tubes (4). The following calculations illustrate how an effectiveness of this heat exchanger was arrived at.

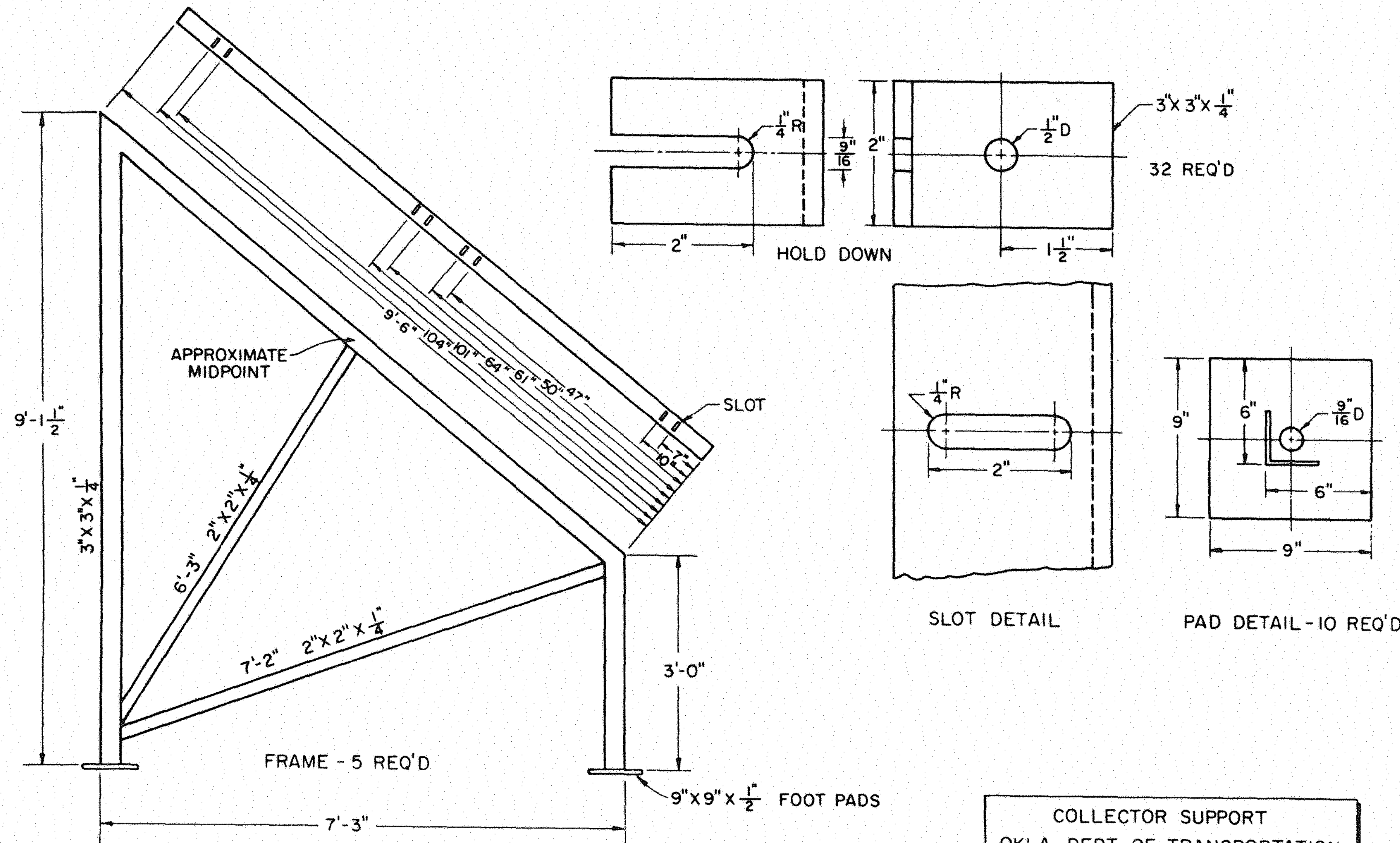
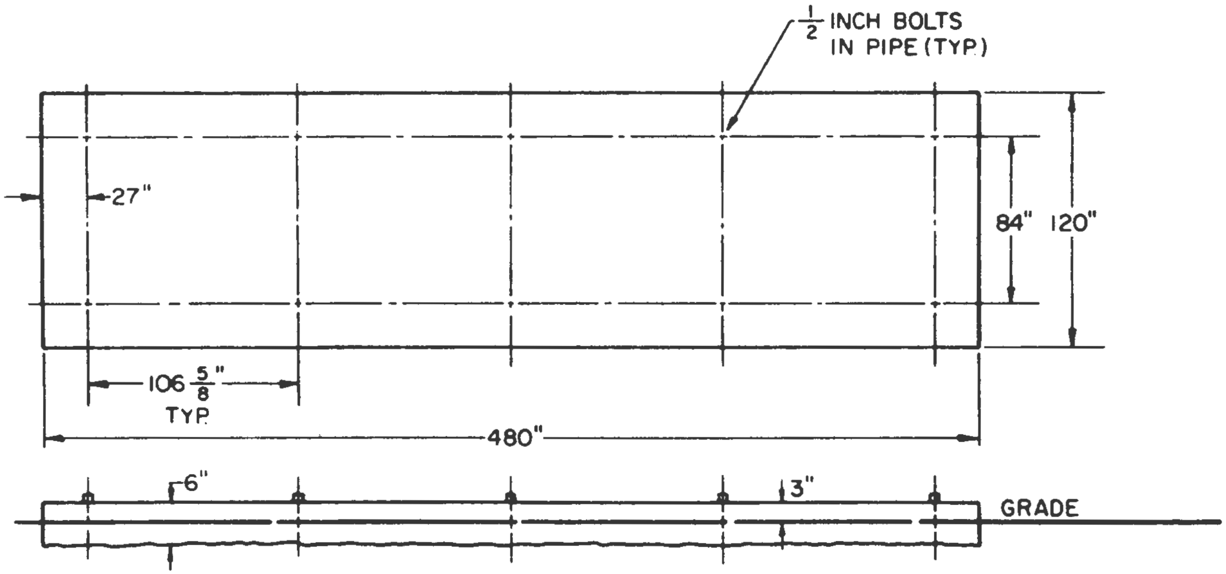


FIGURE 5

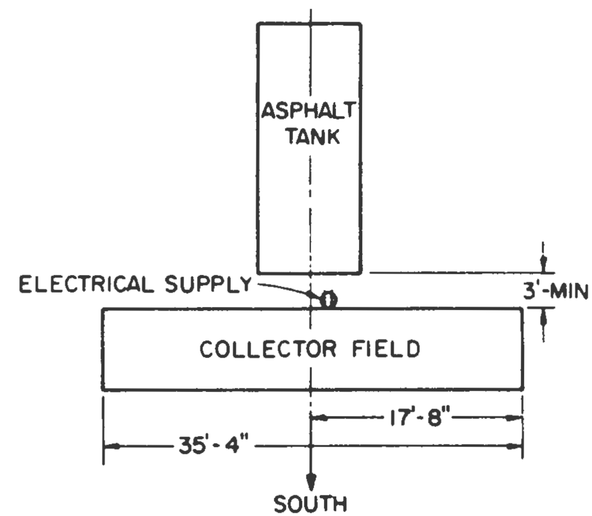
COLLECTOR SUPPORT
OKLA. DEPT. OF TRANSPORTATION
PERRY, OKLA.
2/22/79 J. A. WIEBELT

1 ASPHALT TANK - COLLECTOR FIELD RELATIVE LOCATION - ODOT SET-

2 ELECTRICAL SUPPLY - 220 VOLT - SINGLE PHASE - 5 KW - LOCATED NEAR REAR OF COLLECTORS



COLLECTOR SUPPORT SLAB



PLOT PLAN

ALL DIMENSIONS
± $\frac{1}{2}$ INCH

FIGURE 6

COLLECTOR FOUNDATION
OKLA. DEPT. OF TRANSPORTATION
PERRY, OKLA.
2/28/79 J. A. WIEBELT

An empirical relation (5) was used to calculate the heat transfer coefficient (assumed constant) on the outside surface of the fins:

$$Nu_b = \frac{h_o b}{k} = 0.0649\ [Ra_b(\tfrac{b}{d})]^{0.527}$$

$$b = 0.1/12 \text{ ft}$$

$$d = 2.25/12 \text{ ft}$$

$$Ra_b = Pr\ Gr$$

$$Pr = \frac{\mu c_p}{k}$$

$$Gr = \frac{g\beta \cdot \Delta T b}{\nu^2}$$

From Feldman (6), for MC-series(cutback) asphalts:

$$\rho = 61.7559 - 0.02T$$

assume $T = 200°F$

$$\rho = 57.7559 \ lb_m/ft^3$$

$$k = 0.094\{1 - 11.79875(10^{-4})[\tfrac{5}{9}(T-32)]\}$$

$$k = 0.08365 \text{ BTU/hr ft °F}$$

$$c_p = 0.40925 + 51.875(10^{-5})[\tfrac{5}{9}(T-32)]$$

$$c_p = 0.4577 \text{ BTU/lbm °F}$$

$$\mu = 126.2338(10^{24})T^{(-9.6905)}$$

$$\mu = 6354 \ lb_m/\text{ft hr}$$

also

$$\beta = \frac{1}{v}\left(\frac{\delta v}{\delta T}\right) = \rho\left(\frac{-1}{\rho^2}\right) \frac{\delta\rho}{\delta T} = \frac{-1\ (-0.02)}{57.7559} = .0003463 \ °F^{-1}$$

assume $\Delta T = 30°F$

$$\nu = \frac{\mu}{\rho}$$

$$\nu = 110.0 \ ft^2/hr$$

therefore, $Pr = 34766$

$$Gr = 2.074(10^{-4})$$

$$Ra_b = 7.21$$

$$Nu_b = 0.0356 = \frac{h_o b}{k}$$

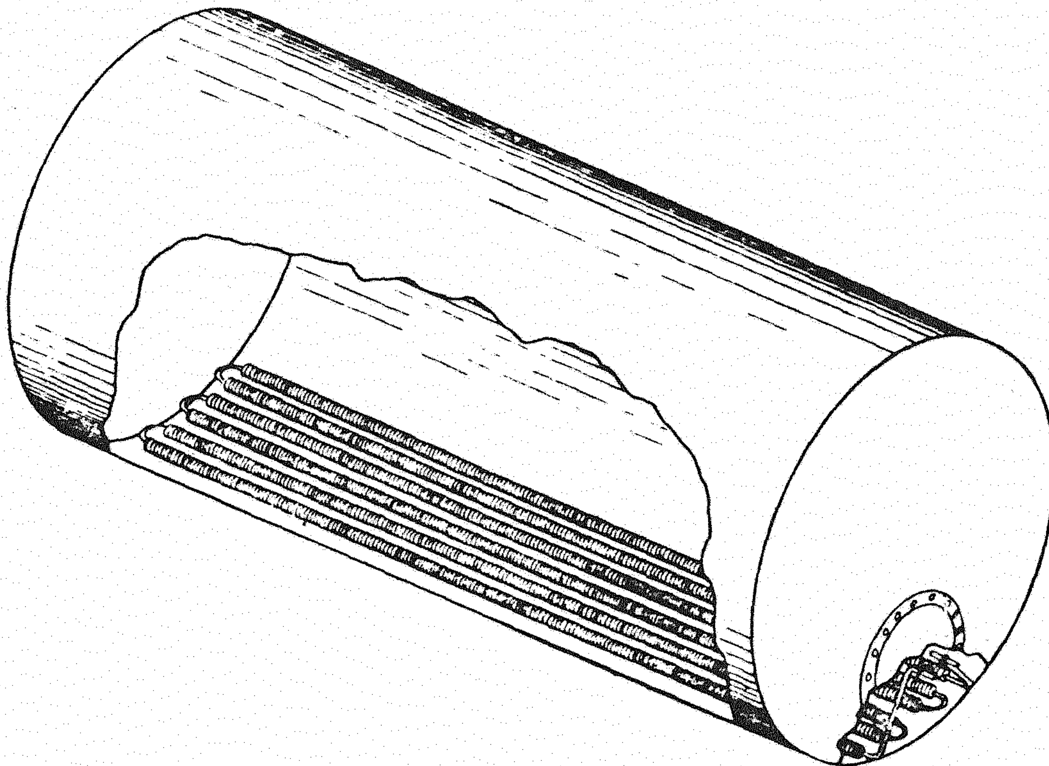


FIGURE 7

Cutaway view of tank and
proposed heat exchanger

The outside heat transfer coefficient for the cutback asphalt was therefore calculated to be h_o = .3576 BTU/hr ft^2 °F.

The design flow rate of each collector is 0.3 gallon per minute. The total flow rate through the heat exchanger is therefore 1.2 gallon per minute. At this flow rate, the $Re_D = \frac{V_\infty D}{\nu}$ = 15075 where V_∞ = 0.460 ft/sec, D = 1/12 ft, and $\nu = 9.15(10^{-3})$ ft /hr. This is fully developed turbulent flow, and the Seider-Tate equation was used for calculating the Nusselt number:

$$Nu_D = \frac{h_i D}{k} = 0.023\ Re_D^{0.8} Pr^{0.33}$$

At 200°F, the water-ethylene glycol Prandtl number is 2.10 and thermal conductivity, k = 0.237 BTU/hr ft °F. The inside heat transfer coefficient is therefore h_i = 184 BTU/hr ft °F.

It was also necessary to calculate a fin efficiency for the heat exchangers. An approximation (7) for circular fins on tubes is given by: (see Figure 8)

$$\eta_f = \frac{\tanh\,(BH_r)}{(BH_r)}$$

$$B = \left[\frac{2\,h_o}{kt}\right]^{1/2}$$

h_o = ·3576 BTU/hr ft² °F.

$k = k_{aluminum}$ = 48 BTU/hr ft °F

t = thickness = 1/32 inch

$B = .6808\ ft^{-1}$

$H_r = r\,[(\rho-1)(1+.35\ \ln\rho)]$

$\rho = \frac{R}{r}$

R = 1.125 inches

r = 0.5 inches

$\rho = 2.25$

$H_r = 0.0669$ ft

$\eta_f = 0.9993$

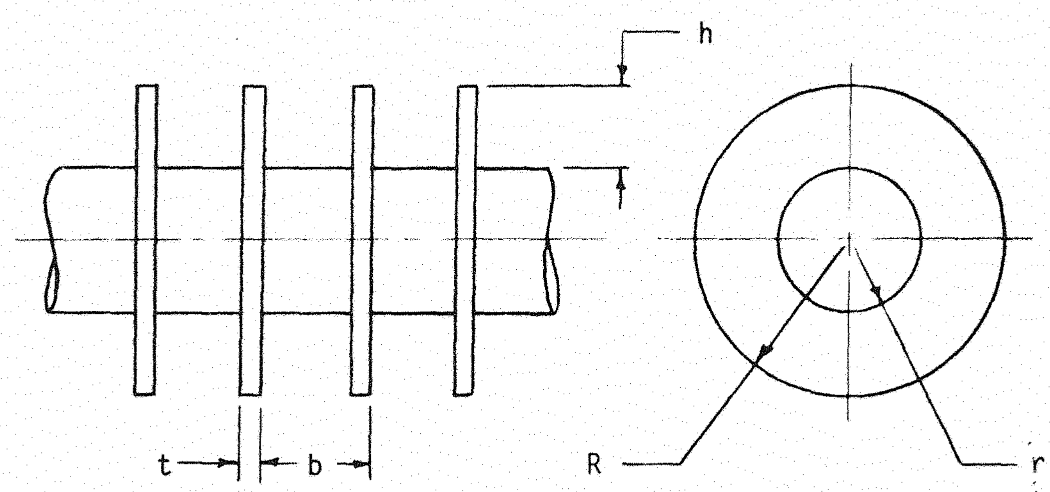


FIGURE 8

Circular finned tube nomenclature

The surface effectiveness is given by:

$$\eta_s = 1 - \frac{A_f}{A_s}(1-\eta_f)$$

A_f = total fin area = 680.7 ft²

A_s = total surface area = 718.4 ft²

$\eta_s = 0.9993$

Then the overall heat transfer coefficient was calculated from the relationship:

$$U = \frac{1}{\frac{1}{h_o \eta_s} + \frac{1}{h_i(A_i/A_s)}}$$

A_i = area inside tube = 33.5 ft²

The calculated overall heat transfer coefficient is thus 0.3431 BTU/hr ft²°F. The NTU method (8) was then used to calculate heat exchanger effectiveness:

$$NTU = \frac{UA_s}{C_{min}} = \frac{UA_s}{c_p \dot{m}} \text{ ethylene glycol-water}$$

$c_p = 0.82$ BTU/lb$_m$ °F

$\dot{m} = 600$ lb$_m$/hr

NTU = 0.5009

For this free-convection case, $\frac{C_{min}}{C_{max}} = 0$ because C_{max} (asphalt) $\to \infty$.

Therefore

$$\text{EFFECTIVENESS} = 1 - e^{(-NTU)}$$

The calculated effectiveness of this particular heat exchanger is therefore 0.394.

CONTROLS

The controls at Perry provide pump control and protection for the collectors. See Figure 9 of the control schematic. When the sun has been shining for 10-20 minutes at an insolation level above 35 BTU/hr ft², the photo sensor has sent a sufficient signal to the controller to energize

PERRY CONTROL SCHEMATIC

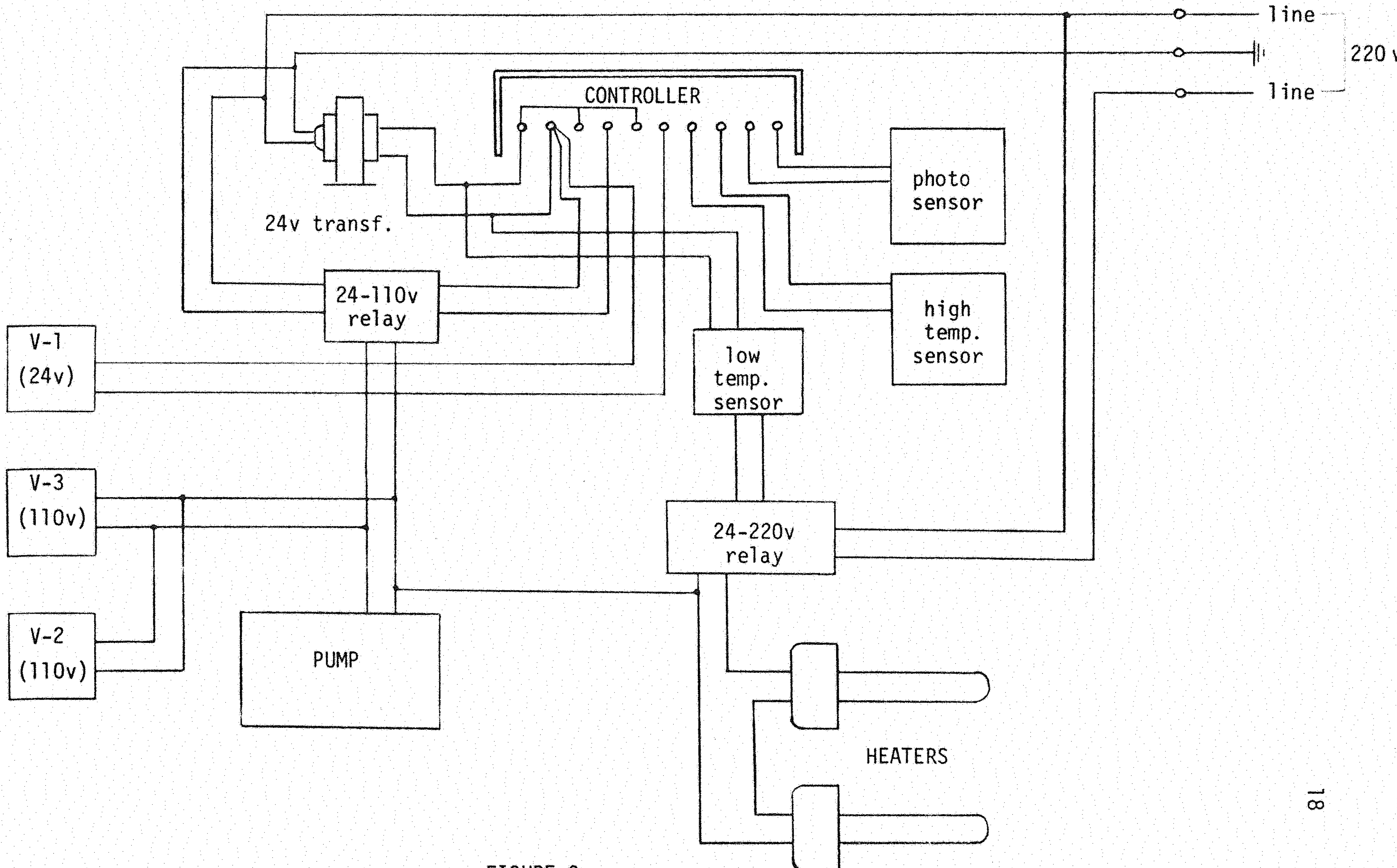


FIGURE 9

the pump. When the pump turns on, valve V-3 opens on the bypass line and valve V-2 closes on the air bleed line. After the pump has run two minutes, the controller energizes valve V-1, closing the bypass line. Flow is then through the collectors until the insolation level is below 35 BTU/hr ft^2. If the high temperature sensor indicates that the collectors are too hot, the controller deenergizes the pump and V-2. This drains down the system, protecting the collectors.

Independent of the controller, the low temperature sensor activates the heaters when the asphalt temperature drops below 185°F. The pump is tied-in to the heaters and must start also. Note that the heaters and collectors may operate together or separately.

The Rho-Sigma controller was provided with the collectors by the manufacturer.

FLUID SYSTEM

Because of the low design flow rate of the Energy Design XE-300 collectors, it was decided to use header pipes at the inlet and outlet. At an overall flow rate of 1.2 gal/min, the highest flow rate in the headers is 0.6 gal/min because the inlet supply and outlet were teed into the middle of the headers and the flow divided. A one-inch pipe was designated for the header pipe, giving a maximum flow rate in either header of less than 0.25 ft/sec. The return and supply line were set at 3/4 inch.

Selection of the pump proved to be unconventional. Not only did the pump selected have to meet head requirements but also temperature requirements. Total pressure drop in the entire fluid system (piping, collectors, and heat exchanger) was estimated to be 27 ft of head. Maximum temperatures expected exceeded 275°F. A Crane 700-series Dynapump was selected because of its high-temperature rating and total head capacity at 1.2 gallons per minute. At this flow rate, a positive suction head greater than 2 ft was determined to always prevent cavitation. At this flow, the pump delivers about 45 ft total head of ethylene glycol and water solution.

CHAPTER III

CONSTRUCTION

Once the more important aspects of a design have been finalized, construction may begin. Often every detail of the final design cannot be made available to the construction crew. A flexible working arrangement is required.

SITE SELECTION AND LAYOUT

At Perry, a site was chosen such that the collectors would never be shaded by other buildings or equipment constructed in the division yard. The collectors face to the direct south, so the support pad was located on the north side of an equipment access road. Guard rails were recommended to avoid accidents.

Consideration of the asphalt tank accessibility was also very important. Locating the tank close to the collectors minimizes heat losses from the piping. At Perry, the tank was located such that trucks could drive under it to load, then proceed without backing. Figure 1 shows two views of the system layout.

The system must be located near outside power lines. Depending on relative fuel costs, accessibility to natural gas lines for the auxiliary heater may also be desirable. Water lines are a convenience but only necessary if some sort of make-up water is required. The system at Perry only requires electricity and was located close to an underground conduit.

COLLECTOR SUPPORT STRUCTURE

As was discussed in the chapter on design the collectors must be placed high enough off the ground to completely drain down in case of power failure. The fluid level in the expansion tank into which they drain must also be high enough to supply adequate suction head for the pump at all times. The dimensions in question are shown in Figure 5.

Figures 5 and 6 show the collector support stands and their spacing. Note the angle of 40°. Originally, it was proposed to support each stand with two concrete piers. ODOT personnel decided, however, that it would be better to pour one large concrete pad to support the entire collector array and related equipment. This decision proved quite beneficial to the appearance and serviceability of the system. Figure 10 is a photo taken after the stands were erected on the support pad.

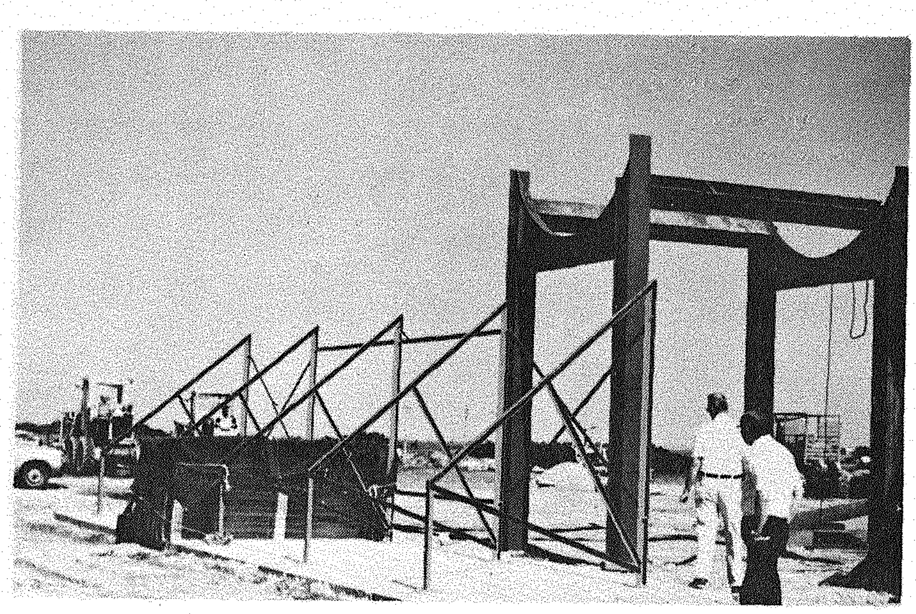


FIGURE 10

Collector support stands. Note concrete pad base.
Photo taken June 1979.

HEAT EXCHANGER

Figure 7 shows the proposed layout of the heat exchanger in the bottom of the asphalt tank. Figure 11 gives the dimensions of the tubes. The tubes were sized for refrigeration systems. To accomodate standard fittings, street ells were used in nonstandard fashion as illustrated in Figure 11.

Before final soldering of the tubes, the entire heat exchanger was assembled on a jig having contour matching that of the bottom of the asphalt tank. This greatly eased final assembly of the tubes inside the tank.

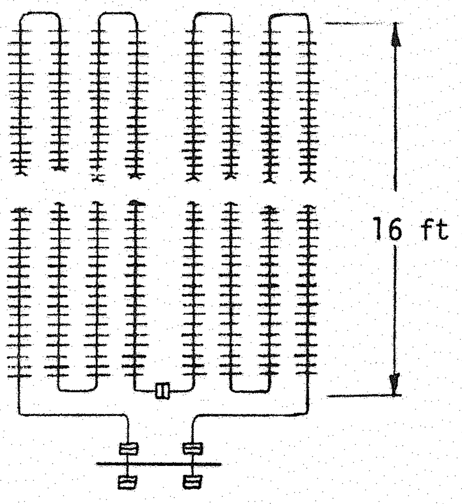
The heat exchanger was completed before the tank was raised into position, so final assembly was simplified. Each half of the heat exchanger was placed in the tank and joined with the other. The lower manway was then fastened to the tank and heat exchanger. The assembler inside the tank got out through the upper manway. Figure 12 is a photo taken during this operation.

COLLECTOR INSTALLATION

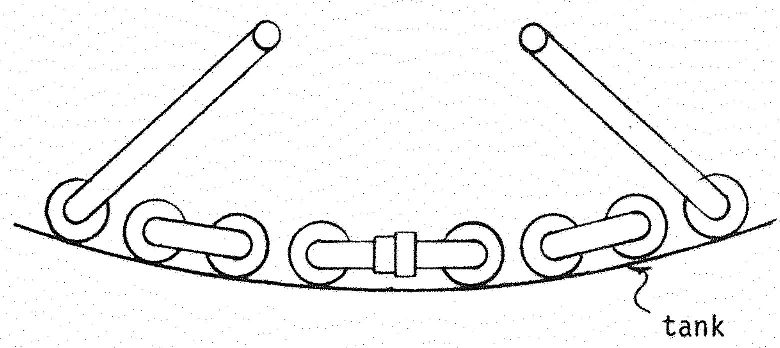
The Energy Design collectors installed at Perry weigh approximately 200 lbs. each. The installation consisted of mounting the ell-shaped brackets on the collectors and then sliding them into position. Figure 13 illustrates this operation.

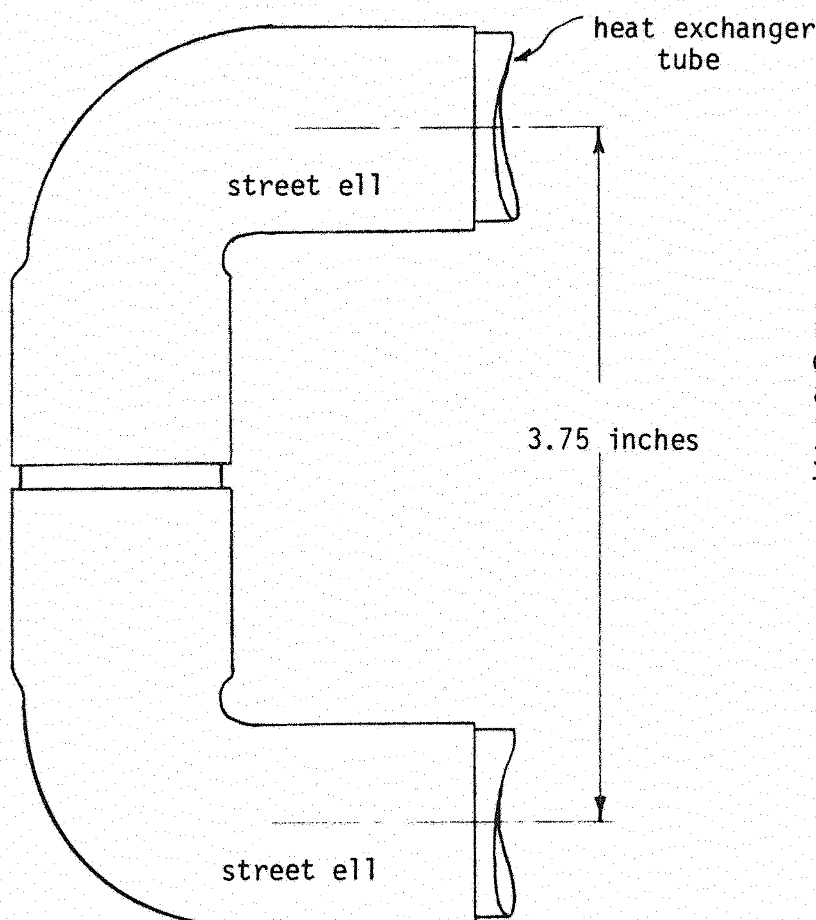
FLUID SYSTEM

The fluid system includes the pump, expansion tank, flowmeter, motorized valves and all of the interconnecting pipes, unions, valves, filter and self-sealing probes. Care was taken to isolate the copper and brass piping and fittings from steel components such as the expansion tank, pump, and asphalt tank. Note that nonconducting unions were used on both sides of the lower manway on the asphalt tank.



plan view schematic





Finned tube heat exchanger, 5/8 inch high aluminum fins, one inch outside diameter admiralty tube, 10 fins per inch, Thermal Engineering, 3545 East 51st Street, Tulsa, Oklahoma 74135.

FIGURE 11

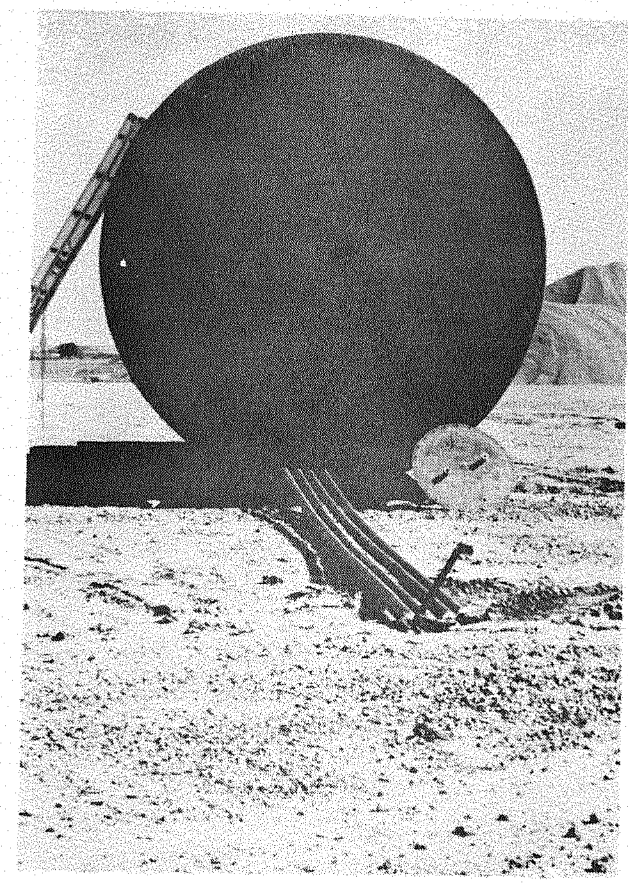


FIGURE 12

Assembling the heat exchanger. One-half is already inside the asphalt tank. Nonconducting unions used to fasten exchanger to manway cover.

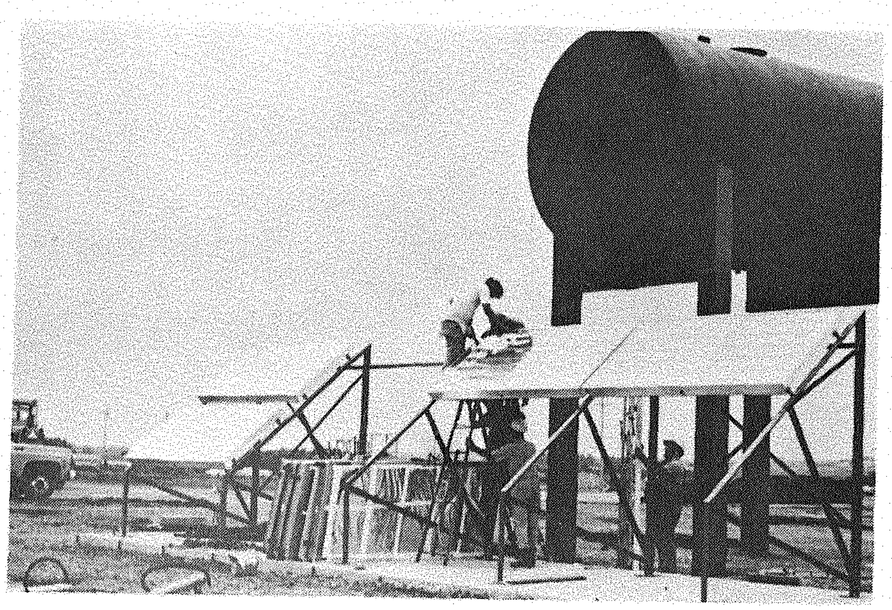


FIGURE 13

Installing the collectors on the stands. This was done on a cloudy day to avoid stagnating dry collectors. To view collector fastening brackets, see figure 3.

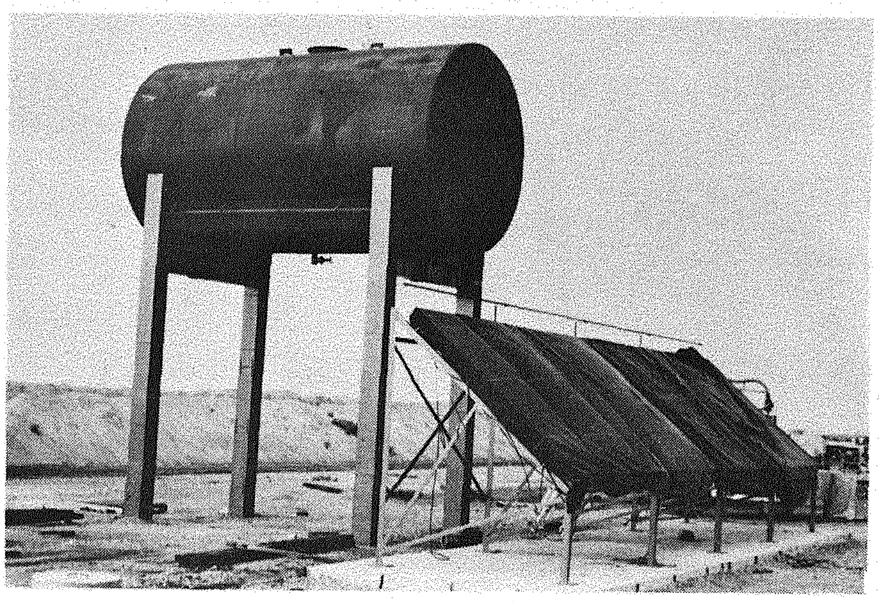


FIGURE 14

Collectors were covered after installation.

All major piping was constructed from type K copper tubes and fittings. Silver solder was used to join all fittings because of the high temperatures expected and for mechanical strength. Only the pipes connected directly to the collectors were of the "soft" type copper tubing. Flare fittings were used to join the pipes to the collectors. Figure 15 shows the basic piping schematic of the system as constructed. Note that 3/4 inch pipe was used for the supply, return, and bypass lines. One inch header pipes connect to the collectors via 1/2 inch soft copper tubes.

The physical arrangement of several components in the system is important. Depending upon the pump used, it is almost always desirable to supply the pump with an adequate nominal positive suction head. A tall, small diameter expansion tank with a known minimum fluid level will do this.

PIPING SCHEMATIC

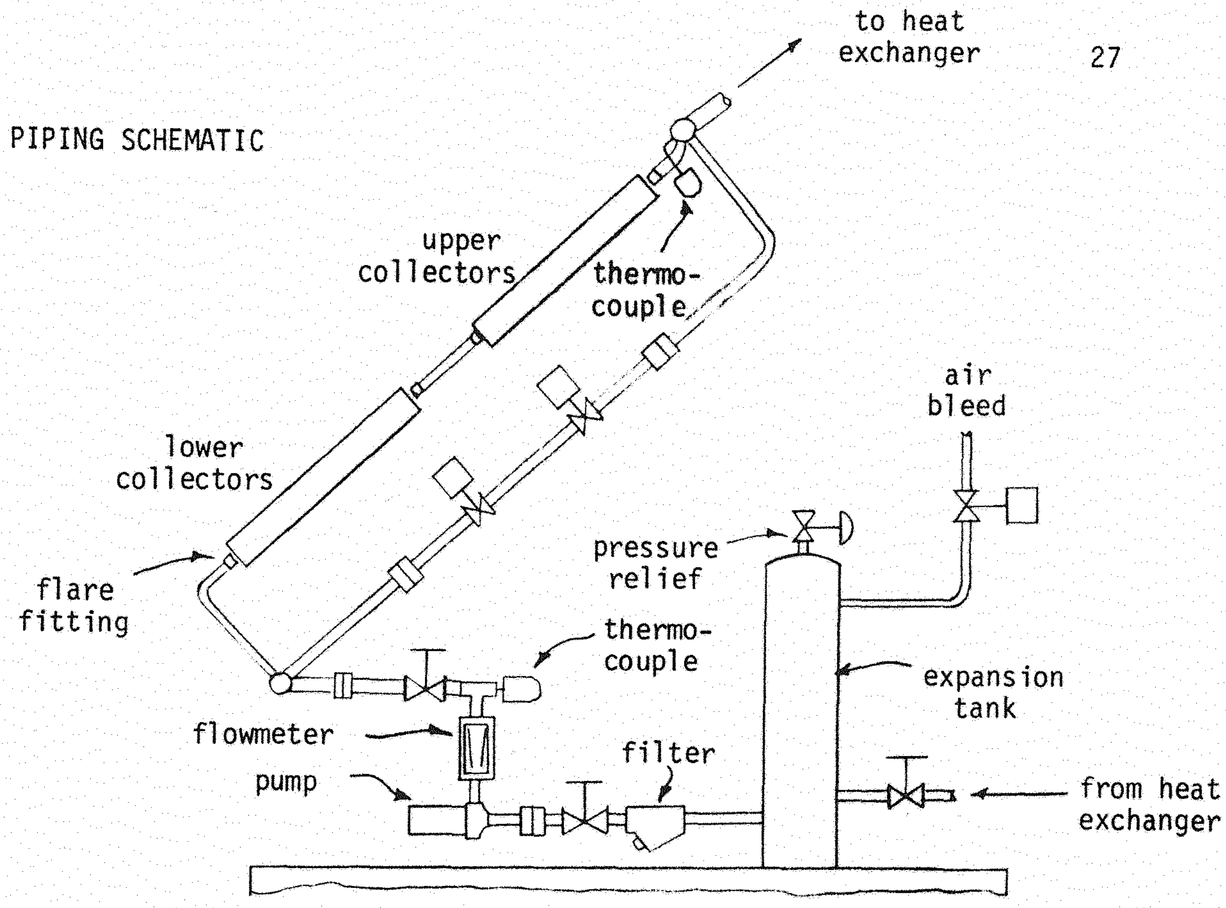


FIGURE 15

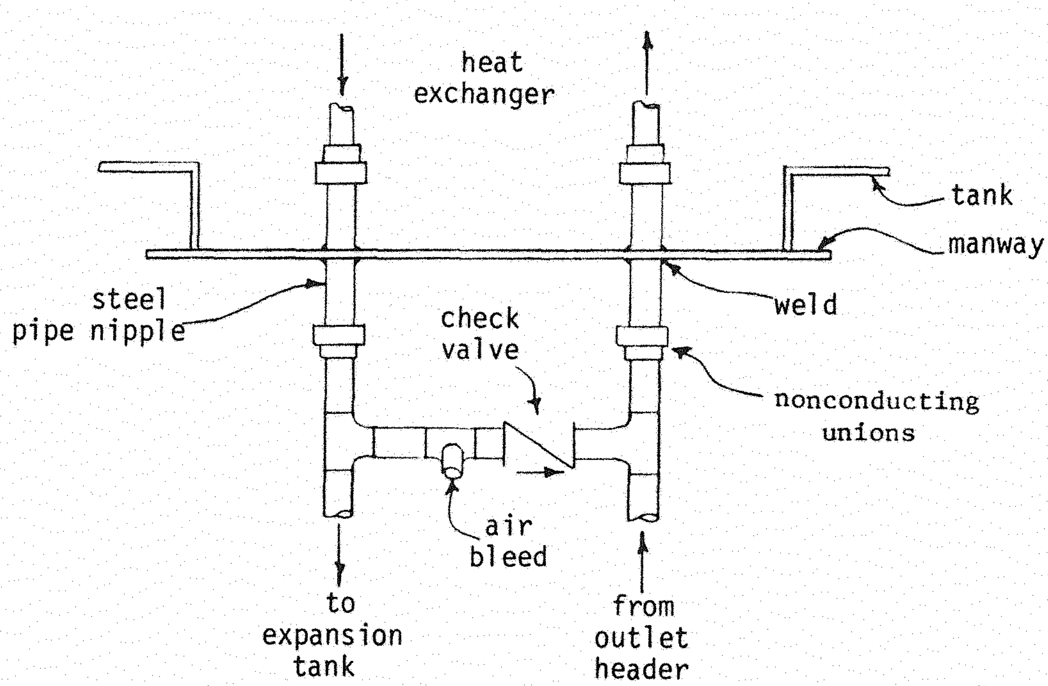


FIGURE 16

Plan view of air bleed scheme.

As discussed in the design chapter, the expansion tank must also be able to accomodate all of the fluid in the collectors. During construction it was realized that the air bleed must accomplish the task of breaking the vacuum at the entrances to the asphalt tank (top of system) such that the expansion tank would not overfill. Figure 16 illustrates the scheme used to accomplish this.

Flow regulation is accomplished by adjusting the gate valve installed near the expansion tank on the return line. The screen filter was installed just upstream of the pump. Unions were used wherever disassembly might be necessary, such as both ends of the bypass line. All pipes were arranged in a fashion to eliminate air pockets.

CONTROLS

Once designed, the installation of the control system was basically one of wiring it up. An electrical enclosure was used to protect the controller, relays, and associated wiring from the elements. A Honeywell circular chart recorder was also installed to record collector inlet, collector outlet, and asphalt tank temperature. In addition to protecting the controls, the enclosure has the advantage of providing a common terminus for all of the external wiring connections. The low temperature sensor was placed remotely in a much smaller electrical enclosure on the tank support structure. This was necessitated by the unavailability of long capillary tubes with these sensors. The sensor is used to switch a 24 volt signal, which switches a 24 to 220 volt relay inside the large electrical enclosure to turn on the auxiliary heaters. Although the sensor used is capable of switching 220 volts, it was deemed undesirable to have nearly 30 feet of high voltage line. Figure 17 shows the arrangement of equipment inside the electrical enclosure.

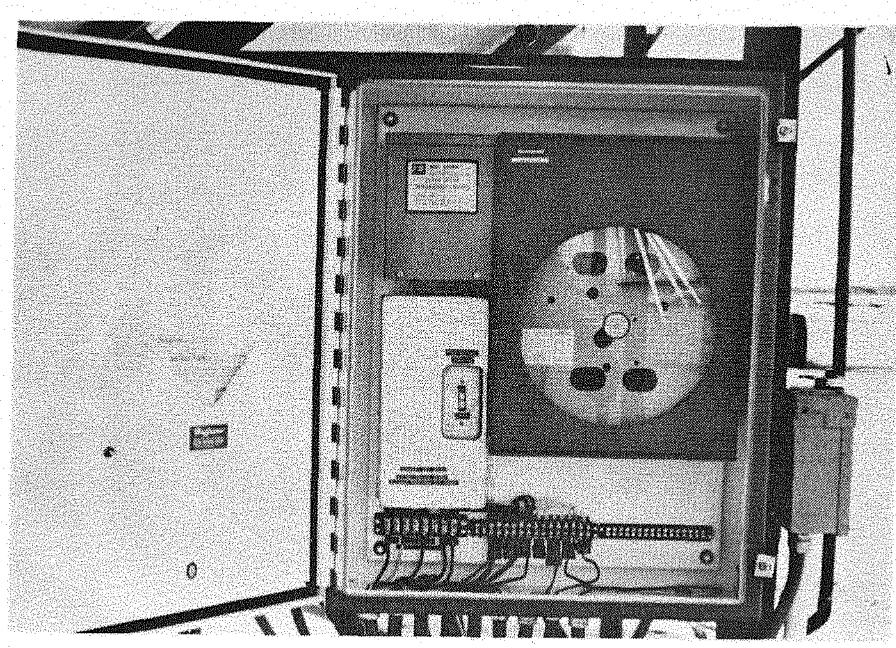


FIGURE 17

Arrangement of equipment within the electrical enclosure. Rho-Sigma integrating controller is at upper left, Honeywell circular chart recorder to the right. Behind metal cover at left are 24-V transformer, 110-V relay, and 220-V relay. Switch on metal cover can be used to override controller to turn on pump manually.

INSULATION

As discussed in the design chapter section dealing with heat loss, proper insulation is the key to making these systems work. At Perry, great care was taken to minimize losses from the asphalt tank. Because the support frame can act as a heat sink, the tank must be isolated from it. ODOT personnel at Perry proposed the idea of using bridge expansion joint rubber to lay between the tank and the support frame. Note that the weight of the tank and asphalt are sufficient to hold it in place with no connection to the frame.

The tank at Perry was insulated with 4 inches of spray-on polyurethane foam. A small air vent was installed at the very top of the tank before insulation. Care was taken to insulate everything--valves, manways, and lifting hooks. Conduit leading to the resistive heaters on the loading valve was insulated right to the tank. After insulation, a weather protective coating of white latex was applied. See Figure 18.

Insulation of the expansion tank was accomplished in a similar fashion, except that the polyurethane insulation mix was poured into a mold around the tank. It was carried out such that the expansion tank has at least two inches of insulation all around it. Before pouring, rigid polyurethane was placed beneath to insulate and raise the tank from the concrete pad. The tank before insulation can be seen in Figure 19, and after insulation in Figure 20.

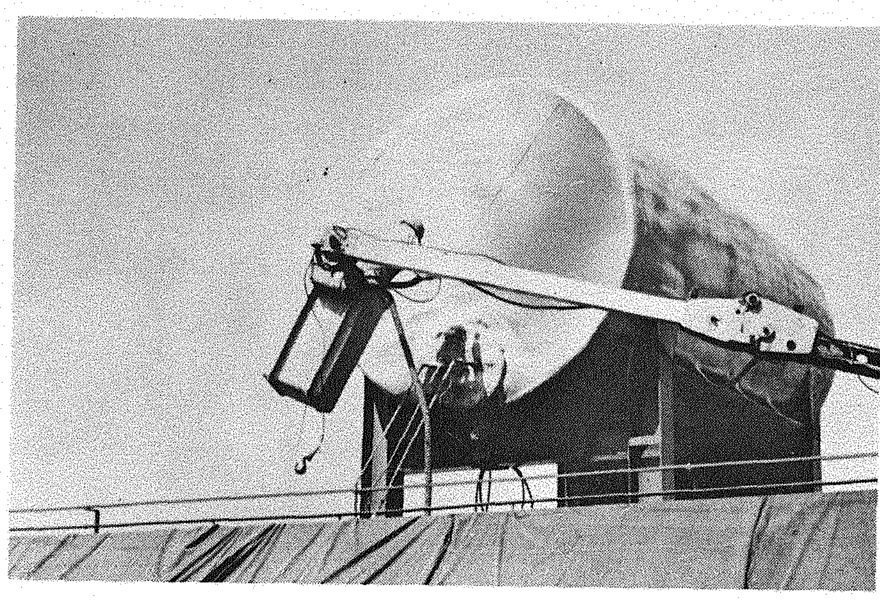


FIGURE 18

Spraying on the polyurethane foam insulation.

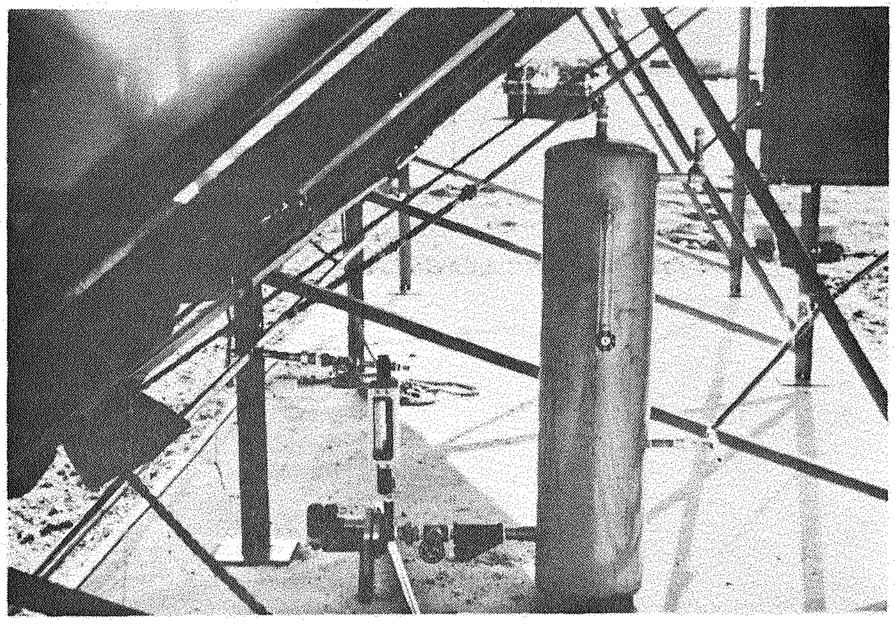


FIGURE 19

View of expansion tank, filter, pump, and flowmeter

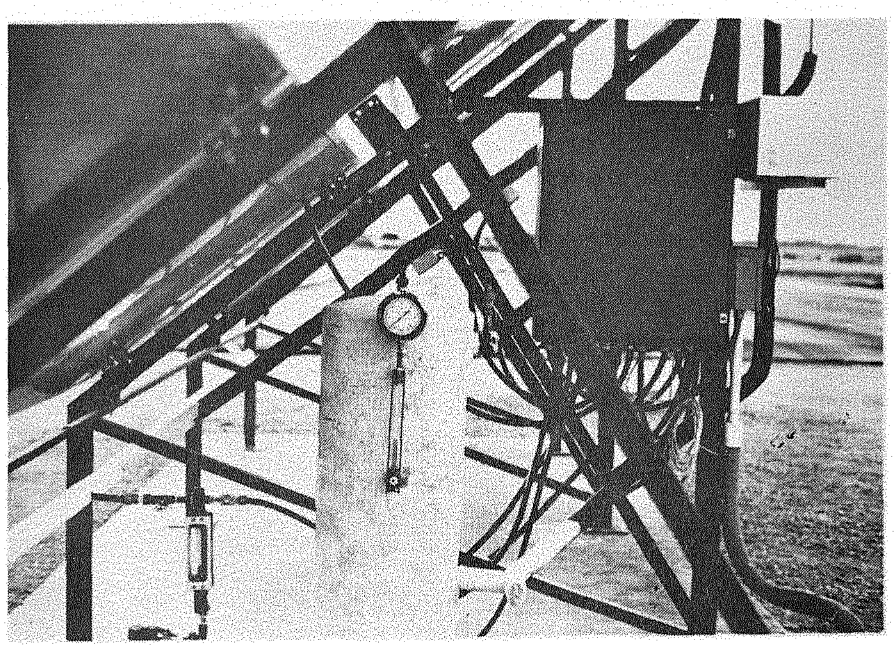


FIGURE 20

Expansion tank after insulation. Note pipe insulation being painted white.

Insulating the copper pipe proved to be quite a challenge. Most solar systems operate at temperatures less than 200°F. These temperatures are easily handled by commercially available rubber foam such as Armaflex or Rubatex.

At Perry, temperatures as high as 300°F were anticipated. Ideally, fibreglass insulation would be installed. This insulation soon appeared as very expensive, difficult to install, and even more difficult to protect from the weather. For these reasons, it was decided to attempt the use of rubber foam on all lines. The insulation was slit, installed, and glued back together. White latex paint was applied to block sunlight.

At the time of this writing, the only uninsulated sections of pipe are the filter, pump face, and flowmeter.

START-UP

At the time of start-up, it was believed that the collectors were very sensitive to stagnation. For this reason, the collectors were covered with canvassing from the time of installation until filled with circulating fluid. See Figure 14.

For initial tests, the system was filled with clean water. Apparently this water caused a leak to appear after a few days of operation by flushing soldering flux away from a joint. The leak was repaired in place--inside the asphalt tank. After passing subsequent tests, the system was drained and refilled with a 50 percent solution of ethylene glycol and water. The pump was left on continuously until the control system was made operational.

The asphalt tank had to be filled with several thousand gallons of water until arrival of the first load of asphalt. This allowed the tarps to be removed from the collectors as system checkouts proceeded. All aspects of the control system were checked for operation.

CHAPTER IV

PRELIMINARY EVALUATION

SYSTEM DUTY CHANGE

During the final stages of construction, it was decided that the asphalt storage system would be used to store emulsified asphalt instead of cutback asphalt. This decision was forced by the rapidly rising cost of kerosene (the solvent in cutback asphalt) and environmental regulations. Unfortunately, this change meant that the Perry system was not to do what it was designed for. Further, this change has caused some performance-related questions to be asked. Additional computer simulations (see Appendix II) indicated that the system may come very close to over-heating the emulsified asphalt in the summer unless a way to expend the additional energy is found. Summarizing the important computer simulation results, based on hourly 1964 weather data for Oklahoma City and a full asphalt tank:

1. The solar collectors should be able to handle the load year round. The auxiliary heater should only be required on rare occasions.
2. The auxiliary heater is oversized.
3. The system may over-heat the asphalt in the summer if there is not enough asphalt in the tank or if the system performance is not altered.

Another important design change during construction was in the amount of insulation applied to the asphalt tank. Four inches of polyurethane foam were applied instead of the recommended five inches. With low-temperature asphalt, this change may prove beneficial.

COMPUTER MODEL

Because so much of a design such as Perry depends on the accuracy of computer simulation, data was taken and compared to the simulation. The ambient temperature and solar insolation were recorded and then provided as data to the computer model. Figure 21 shows the results of this comparison. Due to the short length of data acquisition, it is difficult to make any firm conclusions. It can be seen, however, that the actual temperatures and the simulated temperatures respond to inputs in like fashion. The scale on figure 21 probably exaggerates the differences between the two curves. The maximum difference (3°F) approaches the amount of reading error on the circular charts used to record the data. The thermocouple used to record asphalt temperature may also experience some transients as the asphalt is heated. Convective currents may make the asphalt temperature appear to fluctuate more than the bulk temperature actually does.

THERMAL PERFORMANCE AT PERRY

Before start-up of the Perry system, great care was taken to shield the collectors from stagnation conditions. Once the system was filled with fluid and provided with a load (5000 gallons of cold water in the asphalt tank), an interesting phenomena was observed. Because of the height of the heat exchanger (14 feet above grade), the system will self-circulate at a very slow flow rate. The ethylene glycol solution can be heard boiling inside the collectors (occurs at 285-290°F), but the maximum exit temperature of the collectors never exceeded 300°F even on very sunny days. This occurrence allowed the disabling of the air bleed system such that the system did not have to refill every time the pump started.

The asphalt tank was filled with 8000 gallons of emulsified asphalt early in November, 1979 at an approximate temperature of 140°F. By the end

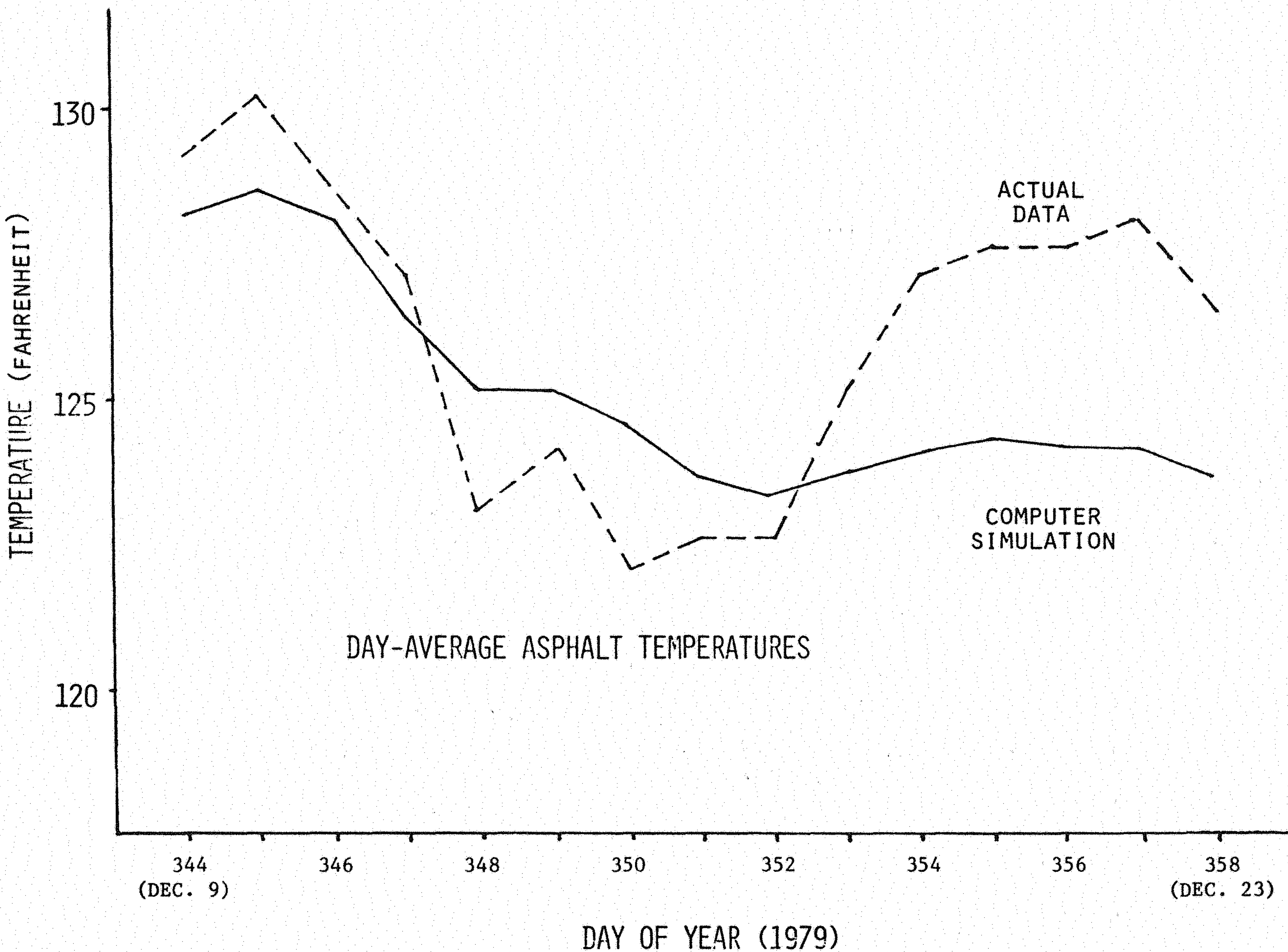

130
125
120
TEMPERATURE (FAHRENHEIT)
ACTUAL
DATA
COMPUTER
SIMULATION
DAY-AVERAGE ASPHALT TEMPERATURES
344
(DEC. 9)
346
348
350
352
354
356
358
(DEC. 23)
DAY OF YEAR (1979)

FIGURE 21

of January the temperature had decreased to almost 100°F. Figure 22 shows the measured asphalt temperatures from December 1, 1979 to January 15, 1980. On a very cold night (10 to 20°F) the bulk asphalt temperature may decrease a full 10°F but will recover almost completely if a sunny day follows. The auxiliary heaters have yet to be used.

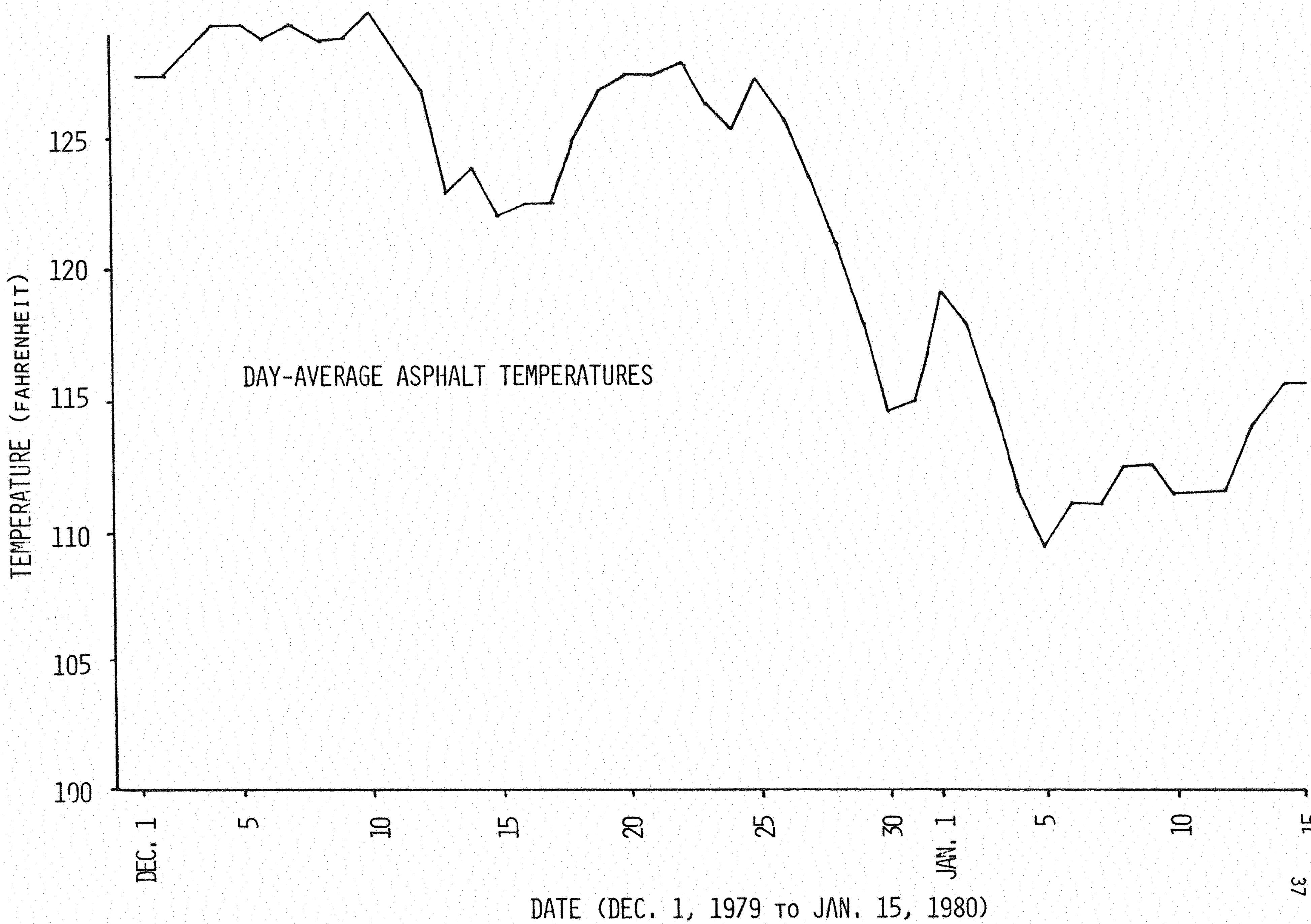
The collectors appear to be working very well. Although performance evaluation is not possible with the amount of data collected so far, it appears that the collectors are performing at least as well as claimed by the manufacturer.

PHYSICAL PROBLEMS

When the fluid piping was installed, the intention was to arrange all headers, supply, and return lines such that there would be no air entrapment. It would have been better to install manual or automatic air bleeds as there have been problems with air pockets in the lines. These unneccessary restrictions may have caused some start-up problems such as getting the air to bleed out of the top of the system.

The 15 psig pressure relief valve was found to leak, allowing the expansion tank pressure to always be the ambient pressure. If the system were operating at higher temperatures, this would have allowed the ethylene glycol mixture to flash--decreasing heat exchanger effectiveness.

Probably the most troublesome problem thus far has been the pump. Occasionally it would not start when activated either by the control system or manually. This was thought to be due to sediment inside the pump casing. It has not been cleaned since start-up. Interestingly, the pump has never failed to start upon demand from the controller when unattended. Perhaps it needs time to warm-up first. The cause of the sediment was improper selection of some of the piping components and incomplete flushing of soldering flux.



DAY-AVERAGE ASPHALT TEMPERATURES
TEMPERATURE (FAHRENHEIT)
125
120
115
110
105
100
DEC. 1
5
10
15
20
25
30
JAN. 1
5
10
15
DATE (DEC. 1, 1979 TO JAN. 15, 1980)

FIGURE 22

Galvanized fittings should not have been used with an ethylene glycol solution. All soldered components should have been completely flushed before connecting to system.

CHAPTER V

RECOMMENDATIONS AND CONCLUSIONS

RECOMMENDED CHANGES

At Perry, the air bleed system should be removed. Any control valve or pipe that can be removed from such a system can potentially improve performance. Control valves may lose their seal or controllers may not properly signal--thus decreasing performance in the long run. It should be remembered that the system at Perry is a bit unusual in having the asphalt tank so high such that it will self-circulate. If the air bleed system is removed, it will be necessary to provide at least a manual air bleed at the top of the system for initial filling. Air bleeds should also be installed at the ends of the upper header pipe such that all potential air pockets may be removed.

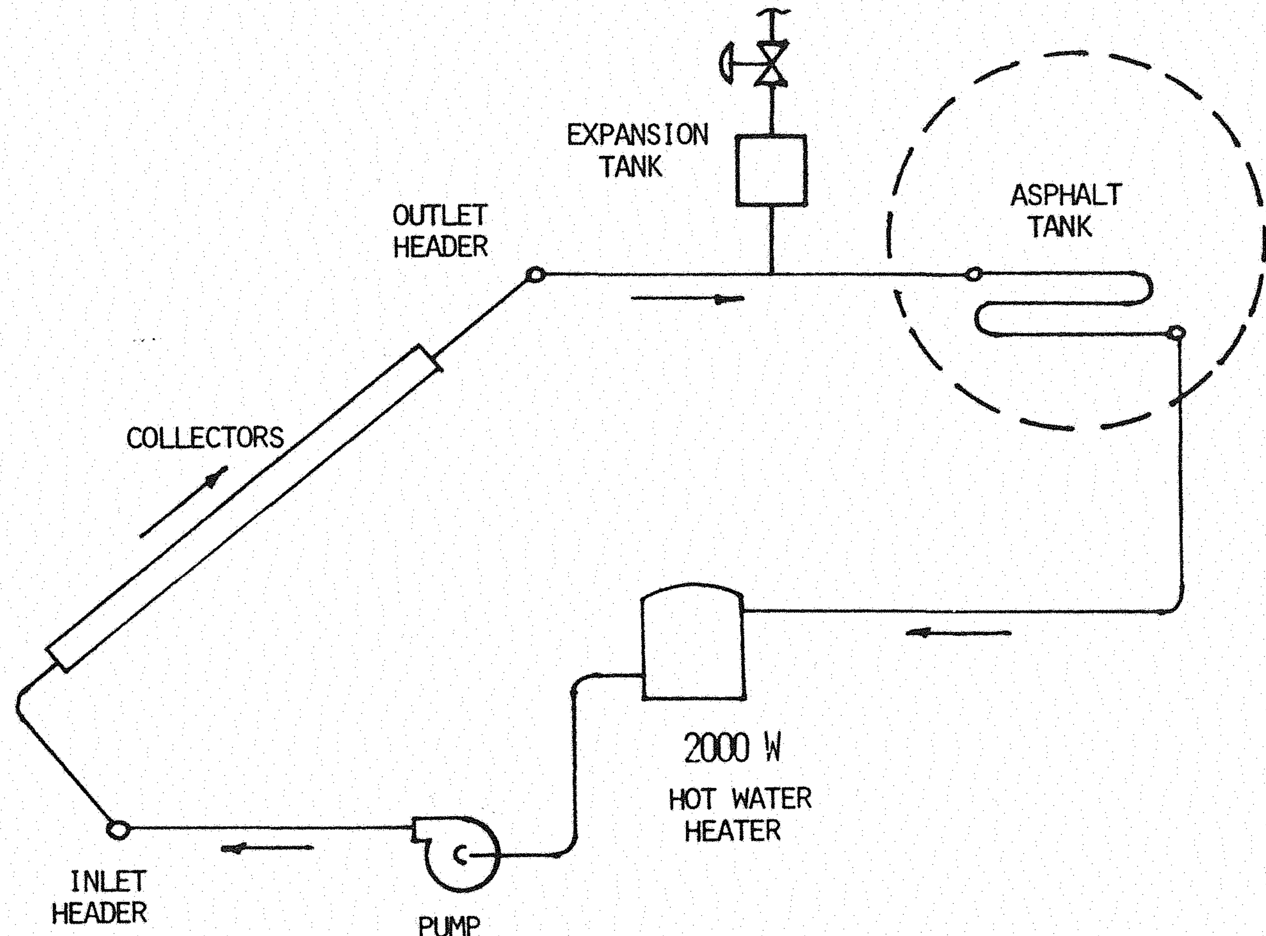
Although the performance at Perry is in no way in need of improvement, the change in asphalt from cutback to emulsion suggests several beneficial changes.

Because of the lower temperatures in the fluid piping system and the rare need for auxiliary heating, the bypass line and associated control valves could also be eliminated. Because of the well-insulated properties of the collectors, not an extremely large amount of heat would be lost by pumping directly through them instead of a bypass line.

At present, the fluid system at Perry contains almost 24 gallons of ethylene glycol-water mixture. Thermal response time is hampered by this large volume of fluid. To decrease the amount of fluid in the system, the expansion tank could be replaced by a smaller volume hot water heater. This water heater could be mounted on a stand so that the pump would always have the necessary suction head. At the same time, the size of the auxiliary heater could be decreased to 2 KW. The hot water heater just described has been installed on asphalt emulsion tanks built for ODOT in Muskogee and Tulsa. If the expansion tank was replaced, it would be necessary to place a much smaller expansion tank at the top of the system. The above steps could probably decrease fluid system volume by 50 percent and gain a corresponding decrease in thermal response time. By eliminating air pockets and possibly leaky control valves and check valves, the self-circulation of the system could be enhanced. Figure 23 illustrates the simplicity these proposed recommendations would impart.

CONCLUSIONS

A solar heated asphalt storage system has been designed, constructed, and is now providing service at the Oklahoma Department of Transportation division maintenance yard in Perry, Oklahoma. The system was designed and constructed to store 10,000 gallons of high-temperature (185-225°F) cutback asphalt with solar energy providing at least 50 percent of the required thermal load. The asphalt tank was instead filled with low-temperature (65-160°F) asphalt emulsion in November, 1979. Solar-derived energy has provided 100 percent of the energy necessary to maintain the asphalt tank within its prescribed temperature limits until February 1, 1980 and should continue to do so. All control systems have operated satisfactorily with no major modifications. Because of the low temperature asphalt being stored, several changes are recommended. They include the removal of the air bleed and bypass line and elimination of the large expansion tank in favor of a smaller commercially-available hot water heater.



PROPOSED CHANGES AT PERRY

FIGURE 23

REFERENCES

1. Ball, Bennett D., The Design of a Solar Heated Asphalt Storage System, Oklahoma State University, Stillwater, Oklahoma, May, 1979.

2. Bennett, C. O. and J. E. Myers, Momentum, Heat and Mass Transfer, Second Edition, New York: McGraw-Hill.

3. Duffie, J. A. and W. A. Beckman, Solar Energy Thermal Processes, New York: John Wiley and Sons, 1974.

4. Parker, J. D., J. A. Wiebelt, and J. B. Henderson, The Use of Solar Energy in the Heating of Asphalt in Storage Tanks, Oklahoma State University, Stillwater, Oklahoma, June 1978.

5. Wiebelt, J. A., J. B. Henderson, and J. D. Parker, Free Convection From the Outside of Radial Fin Tubes, Oklahoma State University, Stillwater, Oklahoma.

6. Feldman, K. T., D. C. Lu, and L. W. Cowley, Computer Aided Design of a Shell and Tube Heat Exchanger for Oil and Asphalt Heating, ASME Paper Number 76-WA/HT-6.

7. Parker, J. D., Extended Surfaces, Notes from Seminar: Augmentation of Heat Transfer Encouraging or Accommodating High Heat Fluxes, Oklahoma State University, February 25-26, 1974.

8. Parker, J. D., J. H. Boggs, and E. F. Blick, Introduction to Fluid Mechanics and Heat Transfer, First Edition, Reading, Massachusetts: Addison-Wesley Publishing Co., Inc., 1974.

APPENDIX I

COMPUTER PROGRAM LISTING

```
C
C   INITIALIZING TEMPERATURES AND PARAMETERS
C
       DATA T1,T2,T3,TTANK/42.,42.,42.,129./
       REAL LAT
       DATA AC,FLOW,CP,CTA,LAT,ICOL/224.,599.4,.85,.6981,.6336,1/
       INTEGER*2 IH,ITDB,ITWB,IV
       REWIND 9
       FL=FLOW
C
C   BEGINNING OF PROGRAM TO CALCULATE PERFORMANCE
C
       WRITE(6,100)
   100 FORMAT(/////)
       WRITE(6,105)
       WRITE(6,115)
       WRITE(6,125)
   105 FORMAT(' ',41X,'INCIDENT',28X,'DAY',23X,'TANK')
   115 FORMAT(' ',10X,'TOTAL',9X,'AUXILIARY',9X,'SOLAR',9X,'SOLAR',14X,'A
      @VERAGE',7X,'PERCENT',4X,'TEMPERATURE')
   125 FORMAT(' ',2X,'DAY',3X,'LOAD (BTU)',2X,'CONTRIBUTION (BTU)',2X,'FL
      @UX (BTU)',2X,'CONTRIBUTION (BTU)',2X,'EFFICIENCY (%)',2X,'SOLAR (%
      @)',3X,'MIN',4X,'MAX',//)
       QTOT=0.
       SOLYR=0.
       AUXYR=0.
       PERYR=0.
       XQYR=0.
       DO 50 IDAY=1,366
       Q=0.0
       SOLSUM=0.0
       AUXSUM=0.0
       XQSUM=0.0
       TM=0.0
       TN=300.
    10 READ(9) IH,ITDB,ITWB,IV,FLUX
       IHOUR=IH
       TA=ITDB
       FLUX=FLUX*221.14
       FLOW=FL
C
C   CALLING SLFX TO CALCULATE FLUX ON PLANE OF COLLECTORS.
C
       CALL SLFX(CTA,LAT,IHOUR,IDAY,FLUX,ICOL,SOLF,FB,FD)
       IF(SOLF.GT.105.) GO TO 17
       QC=0.0
       QAUX=0.0
       IF(TTANK.LT.65.) QAUX=13652.
       GO TO 20
C
C   CALLING EFFI TO CALCULATE COLLECTOR EFFICIENCY
C
    17 CALL EFFI(ICOL,SOLF,TA,T1,T2,FB,FD,EF)
       QC=EF*SOLF*AC
       IF(QC.LE.0.0) QC=0.0
       QAUX=0.0
       IF(TTANK.LT.65.)QAUX=13652.
```

```
      T2=T1+QC/(FLOW*CP)
      T2A=T2+QAUX/(FLOW*CP)
      IF(TTANK.LT.160.) GO TO 30
      XQSUM=XQSUM+QC
      T2=T1
      T2A=T2
      FLOW=0.0
C
C  CALLING TANK TO CALCULATE FLUID OUTLET TEMPERATURE
C  AND NEW TANK TEMPERATURE.
C
   30 CALL TANK(T2A,TTANK,CP,FLOW,QC2,TA,T3,TTANK2)
      TTANK=TTANK2
      T1=T3
C
C
      Q=Q+QC2
      SOLSUM=SOLSUM+SOLF*AC
      AUXSUM=AUXSUM+QAUX
      TMAX=AMAX1(TM,TTANK)
      TM=TMAX
      TMIN=AMIN1(TN,TTANK)
      TN=TMIN
      IF(IHOUR.LT.23) GO TO 10
      QASUM=Q-AUXSUM
      IF(SOLSUM.GT.0.) GO TO 87
      DAYEFF=0.
      GO TO 89
   87 DAYEFF=(QASUM/SOLSUM)*100.
   89 IF(Q.NE.0.) GO TO 93
      PERSUM=0.
      GO TO 95
   93 PERSUM=(QASUM/Q)*100.
   95 QTOT=QTOT+Q
      XQYR=XQYR+XQSUM
      SOLYR=SOLYR+QASUM
      AUXYR=AUXYR+AUXSUM
      PERYR=(SOLYR/QTOT)*100.
      IF(MOD(IDAY,20).NE.1) GO TO 50
      WRITE(6,135) IDAY,Q,AUXSUM,SOLSUM,QASUM,DAYEFF,PERSUM,TMIN,TMAX
  135 FORMAT(' ',1X,I3,4X,E9.2,5X,E9.2,8X,E9.2,6X,E9.2,13X,F5.1,9X,F5.1,
     @5X,F5.1,2X,F5.1)
   50 CONTINUE
      WRITE(6,220)
  220 FORMAT(' ',//,' YEAR TOTALS:',/)
      WRITE(6,222)
  222 FORMAT(' ',5X,'TOTAL',11X,'SOLAR',13X,'AUXILIARY',8X,'PERCENT',8X,
     @'EXTRA')
      WRITE(6,224)
  224 FORMAT(' ',3X,'LOAD (BTU)',2X,'CONTRIBUTION (BTU)',2X,'CONTRIBUTIO
     @N (BTU)',2X,'SOLAR (%)',5X,'SOLAR (BTU)',//)
      WRITE(6,226) QTOT,SOLYR,AUXYR,PERYR,XQYR
  226 FORMAT(' ',3X,E9.2,6X,E9.2,11X,E9.2,9X,F5.1,9X,E9.2)
      REWIND 9
      STOP
      END
C
C
```

```
C
C
C
      SUBROUTINE SLFX(CTA,LAT,IHOUR,IDAY,FLUX,ICOL,SOLF,FB,FD)
C
C  THE PURPOSE OF SLFX IS TO CALCULATE THE INCIDENT SOLAR
C  FLUX ON THE PLANE OF THE COLLECTORS AND RETURN AS SOLF.
C
C    CTA      COLLECTOR TILT ANGLE
C    LAT      LATITUDE
C    IHOUR    HOUR OF DAY WHERE 0 IS MIDNIGHT AND 23 IS 11 P.M.
C    IDAY     DAY OF YEAR WHERE JAN. 1 IS 1.
C    FLUX     HORIZONTAL SURFACE FLUX
C    ICOL     TYPE OF COLLECTOR:
C               1)  ENERGY DESIGN XE-300
C
C
C
C    SOLF     RESULTING FLUX ON PLANE OF COLLECTOR
C    FB
C    FD
C
C
      REAL LAT
      IF(FLUX.EQ.0.0) GO TO 700
      SINS=SIN(CTA)
      COSS=COS(CTA)
      SINL=SIN(LAT)
      COSL=COS(LAT)
      ANG=(284.+FLOAT(IDAY))/365.
      ANG=6.2832*ANG
      SOLD=0.4093*SIN(ANG)
      SIND=SIN(SOLD)
      COSD=COS(SOLD)
  120 HANG=FLOAT(IHOUR-12)-0.5
C  HOUR ANGLE
      HANG=0.26185*HANG
C  ZENITH ANGLES
      COSH=COS(HANG)
      SINH=SIN(HANG)
      COSA=1.0
      PA=SIND*SINL*COSS
      PB=SIND*COSL*SINS*COSA
      PC=COSD*COSL*COSS*COSH
      PD=COSD*SINL*SINS*COSA*COSH
      PE=0.0
      COSZ=SIND*SINL+COSD*COSL*COSH
      IF(COSZ.LT.0.1) COSZ=0.1
      Z=ARCOS(COSZ)
      SINZ=SIN(Z)
      SINPH=(COSD*SINH)/SINZ
      PH=ARSIN(SINPH)
      COSPH=COS(PH)
      TANZ=TAN(Z)
      Y=COSPH*TANZ
      ANGLE=ATAN(Y)
      ANGLE1=CTA+0.6283
      ANGLE2=CTA-0.6283
```

```
      QEXT=429.*COSZ
C  CLOUDINESS INDEX
      CI=FLUX/QEXT
      IF(CI.GT.0.75) CI=0.75
C  LINEAR APPROXIMATION OF DIFFUSE TO DIRECT SOLAR INSOLATION
      R=1.0045+((2.6313*CI-3.5227)*CI+0.04349)*CI
      IF(R.LT.0.0) R=0.0
      DFSR=FLUX*R
      DSR=FLUX-DFSR
      IF(ANGLE.GT.ANGLE1.OR.ANGLE.LT.ANGLE2) DSR=0.0
C  CALCULATION OF SOLAR FLUX
      BEAM=DSR*COSPH/COSZ
C  DIFFUSE FACTOR
      DF=0.5*(1.0+COS(CTA))
      SOLF=BEAM+DF*DFSR
      FB=BEAM/SOLF
      FD=DF*DFSR/SOLF
      GO TO 801
  700 SOLF=0.0
      FB=0.0
      FD=0.0
  801 CONTINUE
      RETURN
      END
C
C
C
C
      SUBROUTINE EFFI(ICOL,SOLF,TA,T1,T2,FB,FD,EF)
C
C  THE PURPOSE OF EFFI IS TO CALCULATE THE EFFICIENCY OF
C  COLLECTORS GIVEN THE INCIDENT INSOLATION.
C
C    EPS    ABSORBER EMITTANCE
C    SOLF   SOLAR INSOLATION (BTU/HR*FT**2)
C    FB     FRACTION OF BEAM RADIATION
C    FD     FRACTION OF DIFFUSE RADIATION
C    TAU    TRANSMITTANCE OF COVER SYSTEM
C    TFL    AVE TEMPERATURE THROUGH COLLECTOR
C    TA     AMBIENT TEMPERATURE (FAHRENHEIT)
C    ICOL   COLLECTOR TYPE
C             1) ENERGY DESIGN XE-300
C
C    T1     INLET TEMPERATURE (FAHRENHEIT)
C    T2     OUTLET TEMPERATURE (FAHRENHEIT)
C
C
      TA=TA+460.
      TFL=(T1+T2)/2.+460.
C
      TAU=0.8
      EPS=.035+(TFL-560.)/12000.
C
      A=SOLF*.68*TAU*(1.34*FB+FD)
      B=1.62E-9*EPS*(TFL**4-TA**4)
      C=0.4*(TFL-TA)
C
      EF=(A-B-C)/(1.34*SOLF)
```

```
      IF(EF.LE.0.0) EF=0.0
      TA=TA-460.
      RETURN
      END
C
C
      SUBROUTINE TANK(T2,TTANK,CP,FLOW,QC,TA,T3,TTANK2)
C
C  THE PURPOSE OF TANK IS TO CALCULATE THE THERMAL RESPONSE
C  OF A SOLAR HEATED ASPHALT STORAGE TANK.
C
C     T2       INLET FLUID TEMPERATURE (F)
C     TTANK    PRESENT TEMPERATURE OF TANK (F)
C     CP       SPECIFIC HEAT OF HEATING FLUID (BTU/LBM*F)
C     FLOW     FLOW RATE OF FLUID (LBM/HR)
C     QC       HEAT INPUT BY FLUID (BTU/HR)
C     T3       OUTLET FLUID TEMPERATURE (F)   (OUTPUT)
C     TTANK2 NEW TANK TEMPERATURE (F)   (OUTPUT)
C
C
      US=.060
      AS=806.
      UAHX=1507.
      AMCPS=61600.
C
      QLOSS=US*AS*(TTANK-TA)
      IF(FLOW.GT.0.0) GO TO 20
      QT=-QLOSS
      T3=T2
      GO TO 30
   20 QT=QC-QLOSS
C
      EFEC=1.-EXP(-UAHX/(FLOW*CP))
      T3=T2-((T2-TTANK)*EFEC)
C
   30 TTANK2=TTANK+QT/AMCPS
      RETURN
      END
C
C
END OF DATA
```

APPENDIX II

COMPUTER PROGRAM OUTPUT

OUTPUT ONE

AS BUILT

FULL TANK

DAY	TOTAL LOAD (BTU)	AUXILIARY CONTRIBUTION (BTU)	INCIDENT SOLAR FLUX (BTU)	SOLAR CONTRIBUTION (BTU)	DAY AVERAGE EFFICIENCY (%)	PERCENT SOLAR (%)	TANK TEMPERATURE MIN	TANK TEMPERATURE MAX
1	0.13E+06	0.0	0.37E+06	0.13E+06	35.0	100.0	128.3	129.9
21	0.12E+06	0.0	0.34E+06	0.12E+06	35.0	100.0	139.5	141.0
41	0.19E+06	0.0	0.52E+06	0.19E+06	37.0	100.0	139.8	142.3
61	0.18E+06	0.0	0.49E+06	0.18E+06	36.1	100.0	141.1	143.4
81	0.14E+06	0.0	0.46E+06	0.14E+06	31.3	100.0	138.6	140.4
101	0.14E+06	0.0	0.41E+06	0.14E+06	33.3	100.0	138.5	140.4
121	0.11E+06	0.0	0.37E+06	0.11E+06	29.5	100.0	141.9	143.3
141	0.89E+05	0.0	0.33E+06	0.89E+05	27.3	100.0	137.9	139.0
161	0.72E+05	0.0	0.32E+06	0.72E+05	22.5	100.0	132.9	133.7
181	0.39E+05	0.0	0.26E+06	0.39E+05	15.2	100.0	134.2	134.6
201	0.95E+05	0.0	0.32E+06	0.95E+05	29.4	100.0	138.3	139.6
221	0.82E+05	0.0	0.31E+06	0.82E+05	26.7	100.0	146.9	147.9
241	0.83E+05	0.0	0.30E+06	0.83E+05	27.9	100.0	145.3	146.3
261	0.18E+05	0.0	0.18E+06	0.18E+05	10.0	100.0	148.8	150.0
281	0.15E+06	0.0	0.44E+06	0.15E+06	34.9	100.0	147.2	149.2
301	0.0	0.0	0.93E+05	0.0	0.0	0.0	152.2	153.9
321	0.0	0.0	0.27E+05	0.0	0.0	0.0	147.2	149.0
341	0.14E+06	0.0	0.40E+06	0.14E+06	35.7	100.0	135.9	137.7
361	0.15E+06	0.0	0.42E+06	0.15E+06	34.8	100.0	135.1	136.9

YEAR TOTALS:

TOTAL LOAD (BTU)	SOLAR CONTRIBUTION (BTU)	AUXILIARY CONTRIBUTION (BTU)	PERCENT SOLAR (%)	EXTRA SOLAR (BTU)
0.35E+08	0.35E+08	0.0	100.0	0.0

READY

OUTPUT TWO

AS BUILT

HALF TANK

DAY	TOTAL LOAD (BTU)	AUXILIARY CONTRIBUTION (BTU)	INCIDENT SOLAR FLUX (BTU)	SOLAR CONTRIBUTION (BTU)	DAY AVERAGE EFFICIENCY (%)	PERCENT SOLAR (%)	TANK TEMPERATURE MIN	TANK TEMPERATURE MAX
1	0.13E+06	0.0	0.37E+06	0.13E+06	35.0	100.0	127.6	130.8
21	0.12E+06	0.0	0.34E+06	0.12E+06	34.1	100.0	145.2	148.1
41	0.19E+06	0.0	0.52E+06	0.19E+06	36.7	100.0	141.5	146.4
61	0.18E+06	0.0	0.49E+06	0.18E+06	35.9	100.0	142.0	146.4
81	0.14E+06	0.0	0.46E+06	0.14E+06	31.5	100.0	136.4	140.0
101	0.14E+06	0.0	0.41E+06	0.14E+06	33.3	100.0	137.8	141.5
121	0.11E+06	0.0	0.37E+06	0.11E+06	29.3	100.0	143.2	146.0
141	0.90E+05	0.0	0.33E+06	0.90E+05	27.6	100.0	136.1	138.2
161	0.73E+05	0.0	0.32E+06	0.73E+05	23.0	100.0	128.6	130.4
181	0.39E+05	0.0	0.26E+06	0.39E+05	15.2	100.0	134.0	134.8
201	0.94E+05	0.0	0.32E+06	0.94E+05	29.0	100.0	140.7	143.3
221	0.79E+05	0.0	0.31E+06	0.79E+05	25.9	100.0	152.4	154.4
241	0.82E+05	0.0	0.30E+06	0.82E+05	27.7	100.0	146.4	148.4
261	0.18E+05	0.0	0.18E+06	0.18E+05	10.1	100.0	147.7	150.1
281	0.15E+06	0.0	0.44E+06	0.15E+06	34.9	100.0	146.9	150.9
301	0.0	0.0	0.93E+05	0.0	0.0	0.0	153.3	156.6
321	0.0	0.0	0.27E+05	0.0	0.0	0.0	143.6	147.2
341	0.15E+06	0.0	0.40E+06	0.15E+06	36.7	100.0	127.1	130.9
361	0.15E+06	0.0	0.42E+06	0.15E+06	35.2	100.0	131.1	134.8

YEAR TOTALS:

TOTAL LOAD (BTU)	SOLAR CONTRIBUTION (BTU)	AUXILIARY CONTRIBUTION (BTU)	PERCENT SOLAR (%)	EXTRA SOLAR (BTU)
0.35E+08	0.35E+08	0.0	100.0	0.81E+05

READY

OUTPUT THREE

AS BUILT

FOURTH TANK

DAY	TOTAL LOAD (BTU)	AUXILIARY CONTRIBUTION (BTU)	INCIDENT SOLAR FLUX (BTU)	SOLAR CONTRIBUTION (BTU)	DAY AVERAGE EFFICIENCY (%)	PERCENT SOLAR (%)	TANK TEMPERATURE MIN	TANK TEMPERATURE MAX
1	0.13E+06	0.0	0.37E+06	0.13E+06	35.1	100.0	126.1	132.7
21	0.11E+06	0.0	0.34E+06	0.11E+06	33.3	100.0	150.0	155.3
41	0.19E+06	0.0	0.52E+06	0.19E+06	36.8	100.0	139.2	148.9
61	0.18E+06	0.0	0.49E+06	0.18E+06	35.8	100.0	140.7	149.6
81	0.15E+06	0.0	0.46E+06	0.15E+06	32.1	100.0	130.5	137.9
101	0.14E+06	0.0	0.41E+06	0.14E+06	33.2	100.0	138.1	145.5
121	0.11E+06	0.0	0.37E+06	0.11E+06	29.1	100.0	144.3	149.9
141	0.91E+05	0.0	0.33E+06	0.91E+05	27.7	100.0	134.6	138.9
161	0.74E+05	0.0	0.32E+06	0.74E+05	23.4	100.0	124.7	128.5
181	0.38E+05	0.0	0.26E+06	0.38E+05	14.9	100.0	137.0	138.7
201	0.92E+05	0.0	0.32E+06	0.92E+05	28.6	100.0	143.4	148.4
221	0.79E+05	0.0	0.31E+06	0.79E+05	25.9	100.0	151.6	155.5
241	0.83E+05	0.0	0.30E+06	0.83E+05	27.9	100.0	144.0	148.1
261	0.20E+05	0.0	0.18E+06	0.20E+05	11.2	100.0	138.0	142.0
281	0.15E+06	0.0	0.44E+06	0.15E+06	34.6	100.0	148.0	155.8
301	0.0	0.0	0.93E+05	0.0	0.0	0.0	147.9	154.4
321	0.0	0.0	0.27E+05	0.0	0.0	0.0	136.5	143.3
341	0.15E+06	0.0	0.40E+06	0.15E+06	37.7	100.0	117.7	125.7
361	0.15E+06	0.0	0.42E+06	0.15E+06	35.1	100.0	131.0	138.4

'EAR TOTALS:

TOTAL LOAD (BTU)	SOLAR CONTRIBUTION (BTU)	AUXILIARY CONTRIBUTION (BTU)	PERCENT SOLAR (%)	EXTRA SOLAR (BTU)
0.35E+08	0.35E+08	0.0	100.0	0.54E+06

'EADY

OUTPUT FOUR

COMPUTER SIMULATION
OF ACTUAL DATA
COLLECTED AT PERRY

DAY	TOTAL LOAD (BTU)	AUXILIARY CONTRIBUTION (BTU)	INCIDENT SOLAR FLUX (BTU)	SOLAR CONTRIBUTION (BTU)	DAY AVERAGE EFFICIENCY (%)	PERCENT SOLAR (%)	TANK TEMPERATURE MIN	TANK TEMPERATURE MAX
344	0.11E+06	0.0	0.35E+06	0.11E+06	31.5	100.0	127.1	129.0
345	0.11E+06	0.0	0.34E+06	0.11E+06	31.9	100.0	127.6	129.6
346	0.0	0.0	0.77E+05	0.0	0.0	0.0	126.8	128.9
347	0.0	0.0	0.29E+05	0.0	0.0	0.0	124.3	126.7
348	0.80E+05	0.0	0.27E+06	0.80E+05	29.0	100.0	123.3	124.6
349	0.44E+05	0.0	0.24E+06	0.44E+05	18.3	100.0	122.5	123.7
350	0.44E+05	0.0	0.18E+06	0.44E+05	24.1	100.0	121.3	122.4
351	0.11E+06	0.0	0.36E+06	0.11E+06	31.1	100.0	120.5	122.3
352	0.13E+06	0.0	0.41E+06	0.13E+06	31.3	100.0	120.4	122.6
353	0.13E+06	0.0	0.39E+06	0.13E+06	34.0	100.0	121.0	123.4
354	0.95E+05	0.0	0.31E+06	0.95E+05	30.7	100.0	121.9	123.6
355	0.11E+06	0.0	0.34E+06	0.11E+06	31.9	100.0	122.1	124.0
356	0.57E+05	0.0	0.20E+06	0.57E+05	28.8	100.0	122.4	123.4
357	0.10E+06	0.0	0.30E+06	0.10E+06	34.2	100.0	121.9	124.0
358	0.0	0.0	0.33E+05	0.0	0.0	0.0	121.2	123.2

EAR TOTALS:

TOTAL LOAD (BTU)	SOLAR CONTRIBUTION (BTU)	AUXILIARY CONTRIBUTION (BTU)	PERCENT SOLAR (%)	EXTRA SOLAR (BTU)
0.11E+07	0.11E+07	0.0	100.0	0.0

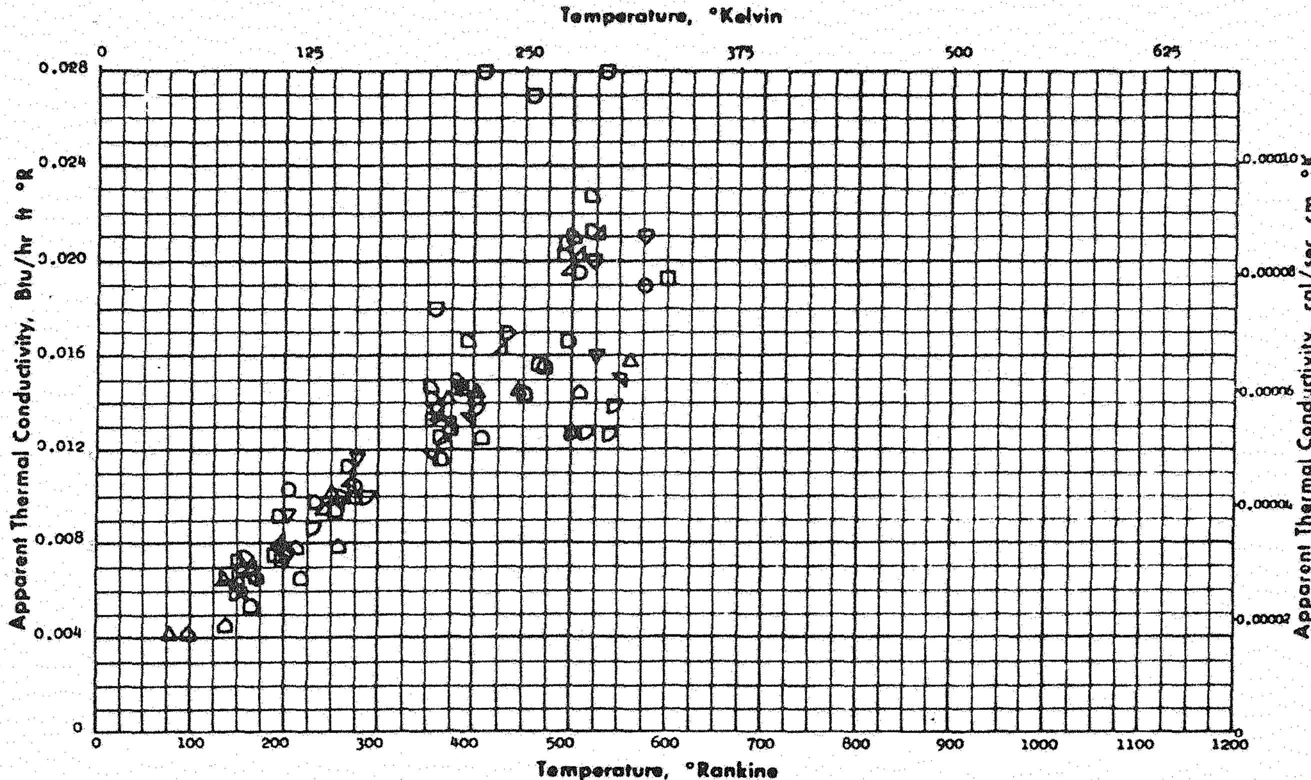
READY

APPENDIX III

INSULATION PROPERTIES DATA

Polyurethane Foam

Apparent Thermal Conductivity

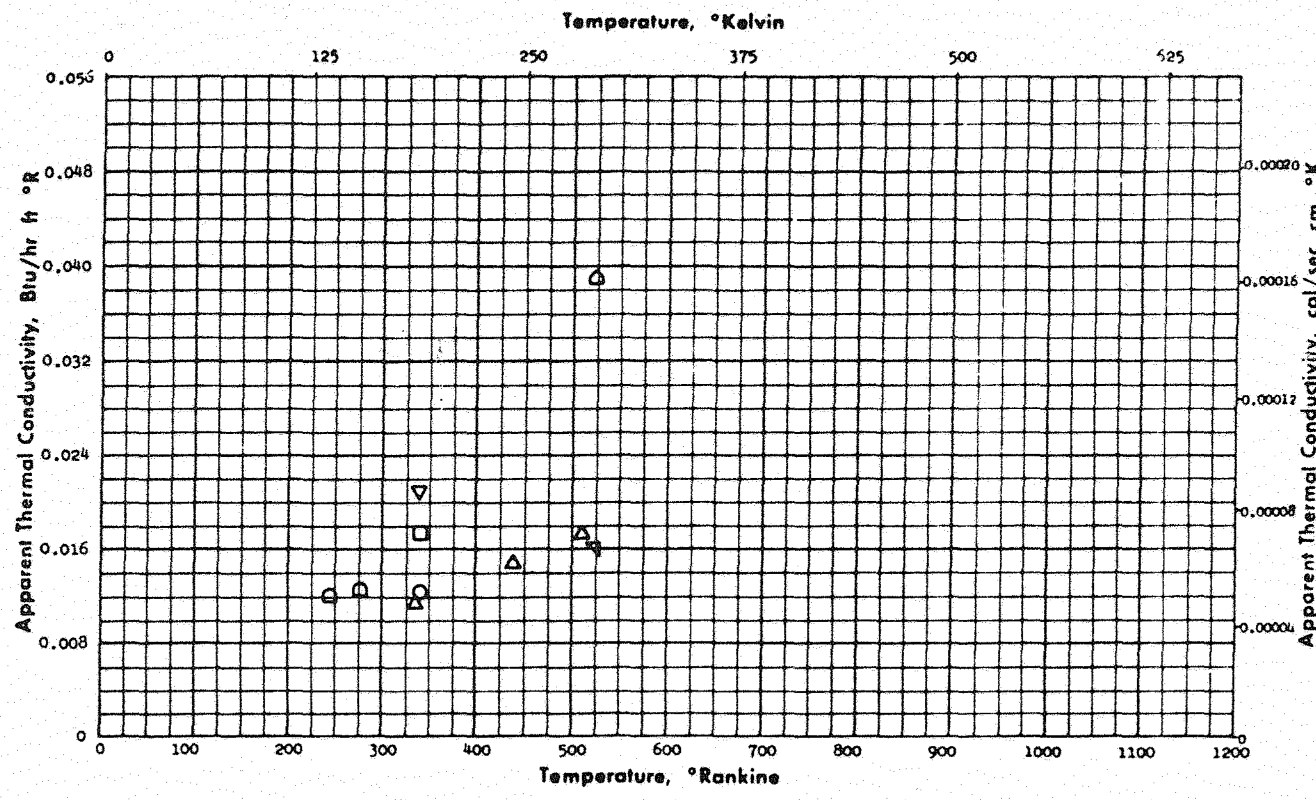


Symbol	Investigator	Ref.	Range, °R	Test Sample	Test Method	Remarks
○	Reichel, R. C.	098	578-206	Polyurethane Foam, sprayed, 8.6 lb/ft^3; "Selectrofoam 6004-6005"	Cylindrical envelope method (Radial heat flow)	Cold temperature: -320°F sealed, atmospheric pressure
□	Reichel, R. C.	098	602-141	Polyurethane Foam, batch mixed; 5.0 lb/ft^3; "Selectrofoam 6004-6005"	Cylindrical envelope method (Radial heat flow)	Cold temperature: -320°F sealed, atmospheric pressure
△	Haskins, J. F., and Hertz, J.	123	80-367	"Stafoam AA-1602"; 2.0 lb/ft^3; freon blown polyester based (American Latex Corporation)	Guarded hot plate method (Twin plate)	Material aged at room temperature for 1-3 months before test; test at atmospheric pressure
▽	Haskins, J. F., and Hertz, J.	123	156-528	"Stafoam AA-402"; 2.0 lb/ft^3; freon blown polyester based (American Latex Corporation)	Guarded hot plate method (Twin plate)	Material aged at room temperature for 1-3 months before test; test at atmospheric pressure
◁	Haskins, J. F., and Hertz, J.	123	156-552	"Polycel-440"; 4.0 lb/ft^3; freon blown polyester based (Polytron Corporation)	Guarded hot plate method (Twin plate)	Material aged at room temperature for 1-3 months before test; test at atmospheric pressure
⌂	Haskins, J. F., and Hertz, J.	123	100-510	"Stafoam AA-3102"; density not given; freon blown polyester based (American Latex Corporation)	Guarded hot plate method (Twin plate)	Material aged at room temperature for 1-3 months before test; test at atmospheric pressure
◘	Haskins, J. F., and Hertz, J.	123	166-474	"ApCO-1414"; density not given; freon blown polyester based (Applied Plastics Div., Hexcel Products, Inc.)	Guarded hot plate method (Twin plate)	Material aged at room temperature for 1-3 months before test; test at atmospheric pressure

continued on next page

Rubber Foam

Apparent Thermal Conductivity



Symbol	Investigator	Ref.	Range, °R	Test Sample	Test Method	Remarks
○	Speil, S.	195	340	Rubber foam; 4.3 lb/ft^3; "Rubatex Board" (Rubatex Division, Great American Industries)	Guarded single plate method	Temperature: -320°F to 80°F (probable), atmospheric pressure
□	Speil, S.	195	340	Rubber foam; 6.3 lb/ft^3; "Aerotube" (Johns-Manville)	Guarded single plate method	Atmospheric pressure
△	Hickman, M. J., and Ratcliffe, E. H.	014	335-510	Foam rubber (cellular ebonite) 4.5 lb/ft^3; 30 cm. x 30 cm. x 3.5 cm. to 5.0 cm. thick	Guarded hot plate method (Twin plate)	Temperature difference from room temperature and -310°F, -103°F, and 32°F
▽	Kropschot, R. H.	069	338	Rubber foam; 5.0 lb/ft^3 (U. S. Rubber)	Guarded hot plate method (Twin plate)	Temperature: 540 to 137°R
◁	Mann, G., and Forsyth, F. G. E.	015	523	New sample, 4.3 lb/ft^3;	Heated probe method, transient heating	
⌂	Mann, G., and Forsyth, F. G. E.	015	523	Old sample, density not given	Heated probe method, transient heating	
◘	Verschoor, J. D.	040	245-277	Expanded rubber board (foam rubber); composition not given; 4.5 lb/ft^3	Guarded hot plate method (Twin plate)	Atmospheric pressure; test specimen dried 24 hr. at 225°F prior to test

APPENDIX IV

ESTIMATED MATERIALS COST

COST ESTIMATES
SOLAR-HEATED ASPHALT STORAGE
PERRY, OKLAHOMA

STORAGE TANK - 10,000 GALLON - WITHOUT STRUCTURE	
SMITH TANK CO., TULSA, OKLA.	$2587.00
TANK INSULATION - 4 INCH POLYURETHANE	
ACI INC., TULSA, OKLA.	3000.00
TANK TOTAL	$5587.00
COLLECTORS - 8-XE-300 WITH SHIPPING	
ENERGY DESIGN CORP., MEMPHIS, TENN.	5642.81
COLLECTOR CONTROLLER - MODIFIED RHO-SIGMA	
ENERGY DESIGN CORP., MEMPHIS, TENN.	200.00
COLLECTOR SUPPORT STRUCTURE	
SHOP BUILT	332.39
COLLECTOR TOTAL	$6175.20
FLOW-METER - BROOKS MODEL 110	
ERNIE GRAVES CO., TULSA, OKLA.	163.74
PUMP - CRANE DYNAPUMP - SERIES 783	
THE GILLILAND CO., TULSA, OKLA.	395.00
MOTORIZED VALVES	
1 - HONEYWELL V8043B 1027	29.43
1 - HONEYWELL V4043A 1259	26.07
1 - HONEYWELL V4043B 1018	25.52
FEDERAL CORP., OKLA. CITY	
FLOW CONTROL TOTAL	$639.76
HEAT EXCHANGER - 8 RUNS - 16 FEET LONG	
1 INCH OD ADMIRALTY TUBES WITH	
5/8 INCH ALUMINUM FINS 10 PER INCH	520.00
THERMAL ENGINEERING, TULSA, OKLA.	
HEAT EXCHANGER TOTAL	$520.00

EXPANSION TANK - 20 GALLON -(12" DIA. X 48" HIGH) 3667N4	
MCMASTER-CARR, CHICAGO, ILL.	$41.18
HEATING UNITS - 2 - 2 KW EA. 3656R13	
MCMASTER-CARR, CHICAGO, ILL.	130.78
WATER TANK GAUGE 3699K11	
MCMASTER-CARR, CHICAGO, ILL.	20.37
THERMOSTAT 3626R12	
MCMASTER-CARR, CHICAGO, ILL.	99.49
RELIEF VALVE 4699K35	14.12
EXPANSION TANK TOTAL	$305.94
HONEYWELL Y-452-X RECORDER WITH 2 - 2T2M15 MEGOPAK THERMOCOUPLE AND 1-1010T-12-1-D THERMOCOUPLE	$1487.50
HONEYWELL INC., OKLA. CITY, OKLA.	
INSTRUMENT BOX - HOFFMAN A-3024-08-LP	
MAC'S ELECTRICAL SUPPLY, TULSA, OKLA.	106.54
RECORDING TOTAL	$1594.04
COPPER PIPE (60'-1", 40'-3/4", 30-1/2")	106.87
COPPER FITTINGS	213.30
ELECTRICAL CONDUIT AND FITTINGS	231.63
ETHYLENE GLYCOL	44.95
ARMAFLEX INSULATION AND PAINT	96.23
MISCELLANEOUS TOTAL	$695.98
TOTAL MATERIALS COST	$15,517.92

APPENDIX V

A DESCRIPTION OF RECENT PROBLEMS

Appendix V

A Description of Recent Problems

On June 10, OSU personnel found that a total of 11 of the evacuated glass tubes had broken in five of the collectors. At the same time, it was discovered that the circulating pump would not function. The system had been shut down for several weeks prior to that time because there was no emulsion in the storage tank. The storage tank had been refilled within the past few days and the system started.

Apparently the tubes broke because the system pump started when the tubes were extremely hot from exposure to the sun. The cool ethylene glycol caused thermal stresses to occur in the glass as it rapidly cooled down the hot tubes.

Normally natural circulation would have kept the tubes at a safe temperature, even with the circulating pump off. The long period of shutdown, however, had allowed the fluid in the collectors to bleed down to the expansion tank and left the collectors dry and capable of extremely high stagnation temperatures.

The system may have been turned on when the collectors were cool, but the pump may have not started at that time. This delayed pump starting had been experienced previously, and may have been a forewarning of the pump failure that finally occurred. It is believed that the pump was faulty from the beginning.

The pump was sent back to the factory, was rebuilt and returned and has been reinstalled. The broken glass tubes have also been replaced.

In addition, a temperature sensor and controller has been installed which will prevent the pump starting if the collectors are at too high a temperature.