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SOLAR POWERED RANKINE CYCLE IRRIGATION PUMP

Final Report

By
William D. Batton
Robert E. Barber

September 1979

Work Performed Under Contract No. AC03-78ET20419

Barber-Nichols Engineering Company
Arvada, Colorado

MASTER

U.S. Department of Energy

Solar Energy

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BARBER-NICHOLS ENGINEERING
6325 W. 55th AVE.
ARVADA, COLORADO 80002

PREPARED FOR THE
DEPARTMENT OF ENERGY
DIVISION OF SOLAR TECHNOLOGY
UNDER CONTRACT DE-AC03-78ET20419

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3.0 ABSTRACT

The use of solar energy as an energy source for a Rankine engine is well known. These solar driven engines can operate any device requiring shaft power including electrical generators, fan blowers, air conditioners, process equipment, and pumps for agricultural crop irrigation.

A new and novel means of combining solar energy with the Rankine engine is to use the collectors as the engine boiler. This boiling-in-collector (BIC) scheme has three major benefits. The first and most significant benefit is the simplicity in the controls that this provides. The operation of the conventional system is very complicated as the temperatures must be controlled by variable flow rates and this system has the problem of determining the proper turn-on conditions while preventing possible over temperaturing of the oil. The boiling-in-collector system, on the other hand, simply heats up to some minimum pressure, turns on and runs, and when sufficient solar is no longer available, shuts down. The second most important benefit is a cost savings in the number of heat exchanges required. This is partially offset by the requirement of using a larger engine to match the peak solar flux (there is no heat storage in this system) but there is a net cost savings. The third benefit is an improvement in engine efficiency as the eliminated heat exchanger process results in a higher turbine inlet temperature.

This final report details the results of test program where a small (288 square feet) collector field was installed and used for boiling-in-the-collector tests with R-113 as a working fluid. Two different types of parabolic trough tracking collectors were purchased and tested. There were two rows (128 sq. ft.) of Del Manufacturing collectors and one row (160 sq. ft.) of Solar Kinetics collectors. All three rows were installed at a 5 degree angle (inclined to the South) oriented North-South and tracking East-West on the roof at Barber-Nichols in Arvada, Colorado, a northwest suburb of Denver. These two types of collectors have distinct differences that made it worthwhile to test each type. A Rankine engine, less turbine expander, was installed and used to complete a solar power system.

The major experimental results are that the collectors did heat the R-113, did provide a vapor suitable for turbine feed, and stable flow did occur under all conditions, thus proving the feasibility of the boiling-in-collector concept. Also, the 5 degree angle performed satisfactorily and is considered a reasonable angle for field use. Some unexpected problems were experienced in testing the collectors. The mirrors on the Del collectors were of poor quality, producing a low efficiency (30%). The exit plumbing on the Solar Kinetics created a vapor pocket resulting in a superheat condition (50°F plus) at the collector exit. This is an undesirable condition. The efficiencies as tested were below the manufacturers' projections (Del 30% vs. 60%, Solar Kinetics 45% vs 60%). In the case of the Del collector this is probably due to the mirrors, but there is no clear cut reason for the discrepancy in the

Solar Kinetics. This discrepancy should be investigated to determine the cause since there is some possibility this is due to the boiling in collector technique and this effect would tend to negate the benefit of the technique. Such an investigation is proposed as part of the first phase of a follow on effort.

The feasibility of the system concept has been demonstrated and areas for improvement have been determined. Methods to resolve these areas are proposed along with a plan to expand the system, complete the engine, and to gather operating experience and performance data for a year's operation. Budget and schedule are provided for this proposed follow on work.

4.0 BACKGROUND

Rankine engines have been used to convert heat energy to work for a number of years. Recently, Rankine engines have been applied to low ($200^{o}F$ to $300^{o}F$) and middle temperature ($400^{o}F$ to $600^{o}F$) ranges where the heat source is either solar, geothermal, or waste heat. These engines use organic working fluids rather than steam because the organics are capable of achieving a reasonable efficiency with a single stage expander (1).

In coupling a Rankine engine with a solar collector system, it is necessary to look at the combined efficiency of the engine and the collector (See "Collector Evaluation and Selection Procedure", Appendix C). The Rankine engine efficiency increases with increasing collector temperature but the collector efficiency drops with increasing collector temperature. Thus, there is some maximum efficiency (i.e., some optimum temperature) for a given collector - engine combination. A part of this coupling that has a significant effect on the efficiency is the temperature drop that takes place in transferring the heat from the solar fluid to the Rankine engine working fluid.

In order to understand this temperature drop, it is necessary to look at the heat exchange process. Figure 4.1 shows this heat exchange process with ABC being the Rankine working fluid and DE the collector loop fluid. Assuming that the engine operates in a temperature region below the critical temperature for the particular working fluid being used, the heating process for the engine working fluid will involve a preheat process A-B, which brings the fluid up to the saturated liquid point, and the boiling process B-C which changes the phase at constant temperature to a saturated vapor. There could be some superheat but in order to maintain a simplified analysis superheat will not be introduced here. The collector fluid normally does not go through a phase change but gives up heat as sensible heat (temperature change). Because the nature of boiling requires some finite temperature difference in order for boiling to take place, the temperature difference PP is some several degrees at a minimum. This in combination with the constant temperature process BC produces a significant temperature drop DC. Thus, a collector exit temperature of D becomes a turbine inlet temperature of C.

When dealing with low temperature engines, even small drops in temperatures can significantly affect the engine performance as the temperature difference between the high side and low side (condenser) is the driving force for the engine. As the condenser rejects heat to the ambient, it is rejecting heat in the 75^{o} to $100^{o}F$ range. Therefore, for a 300^{o} to $400^{o}F$ collector system, the delta T driving force for the engine is 200^{o} to $300^{o}F$. Thus, a $20^{o}F$ drop in the heat exchanger interfacing the collectors to the engine is 6 to 10 percent loss. Of course, there is also an economic consideration as the exchanger in question has some dollar value which is included in the engine cost.

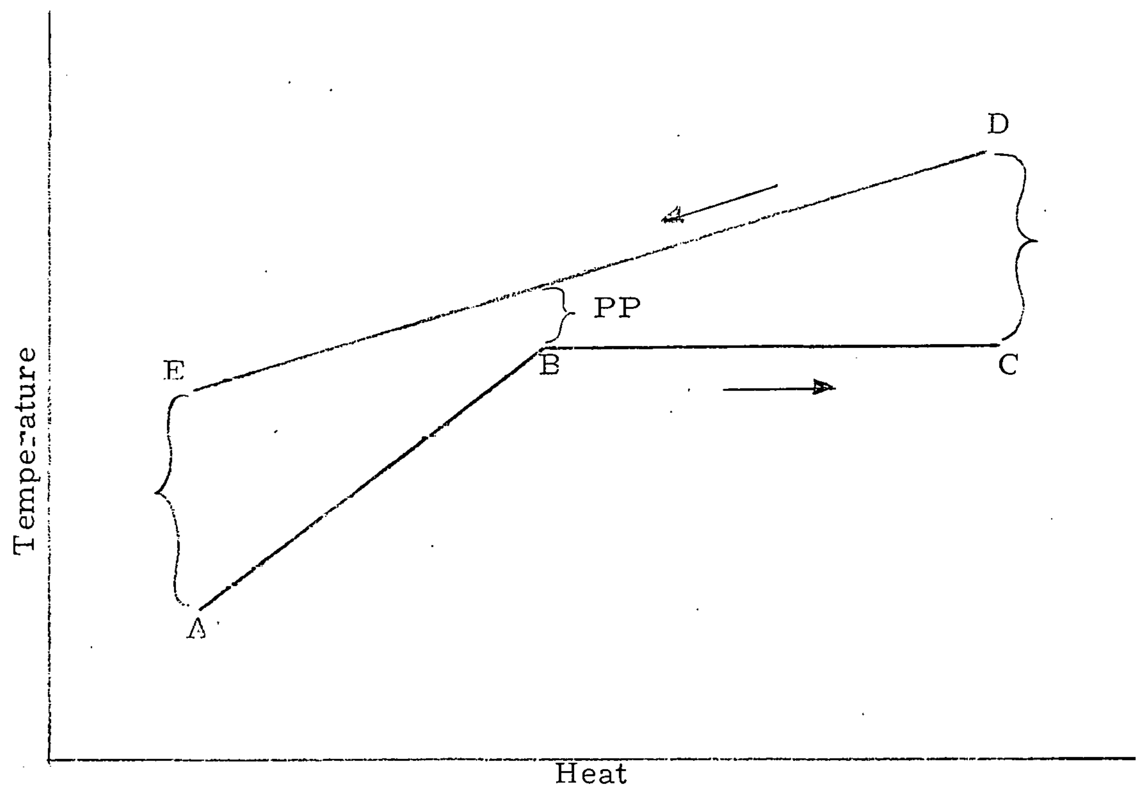


Figure 4.1 Collector - Engine Heat Exchange

The boil-in-collector (BIC) concept eliminates this heat exchanger and allows the engine working fluid to receive heat directly in the collector. Thus, the collector exit temperature (D in figure 4.1) is now the turbine inlet temperature.

The boil-in-collector system is the alternative to the conventional system as previsouly described in this section. The BIC system is substantially smaller than the conventional system due to the smaller number of heat exchangers required. This is dramatized by the artist concept in drawings of the BIC system in Figure 4.2 and of the conventional system in Figure 4.3. These artists conceptions are drawn to scale and use basically the same condenser and regenerator. The power output of both engines is 25 hp.

The conventional system uses a heat transfer oil in the collectors that in turn transfers the heat to the Rankine engine working fluid through a heat exchanger. Such a system is under test by Sandia at Willard, New Mexico. The Rankine engine for that application is a 25-horsepower unit built by Barber-Nichols. The use of the heat transfer oil makes it possible to

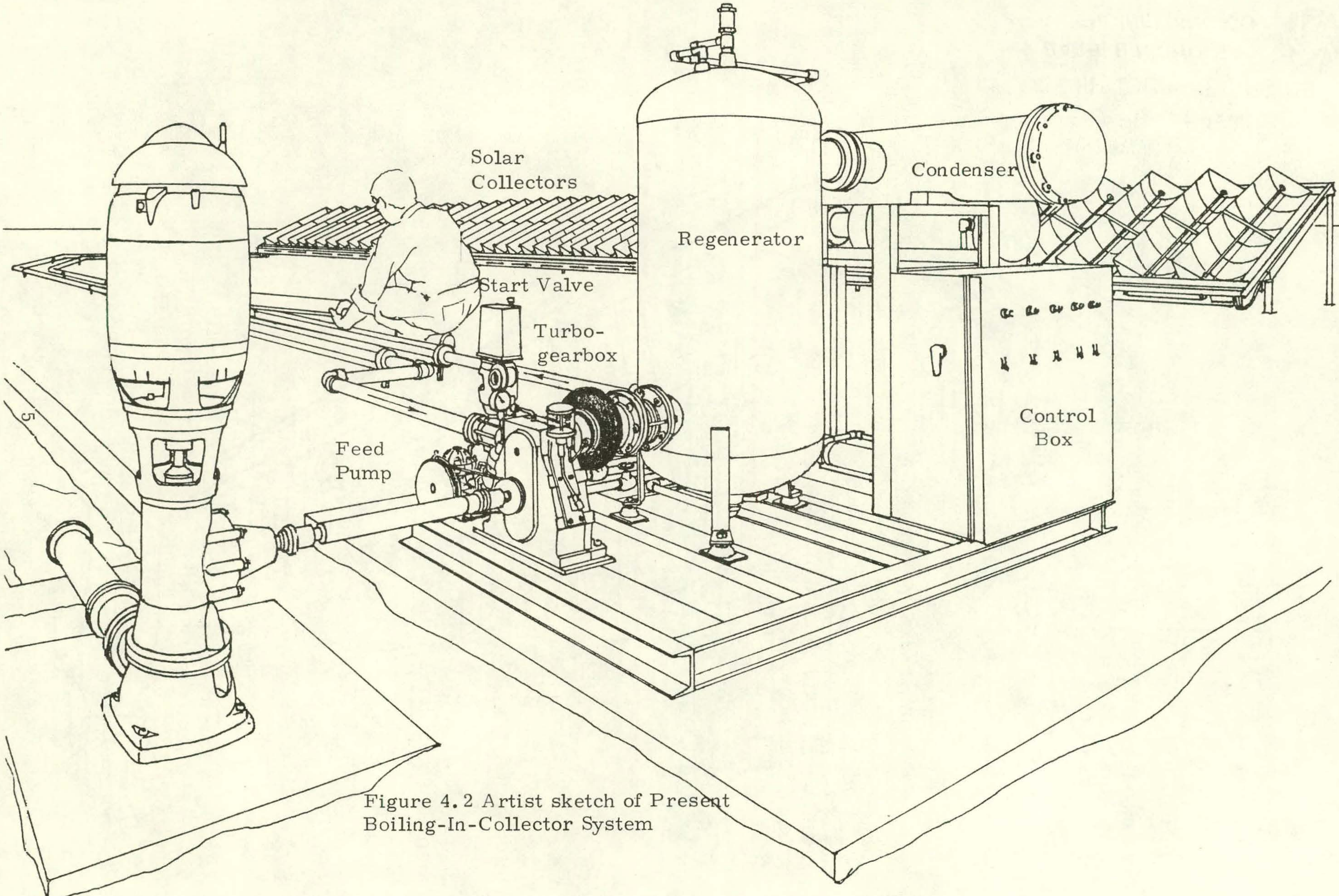


Figure 4. 2 Artist sketch of Present
Boiling-In-Collector System

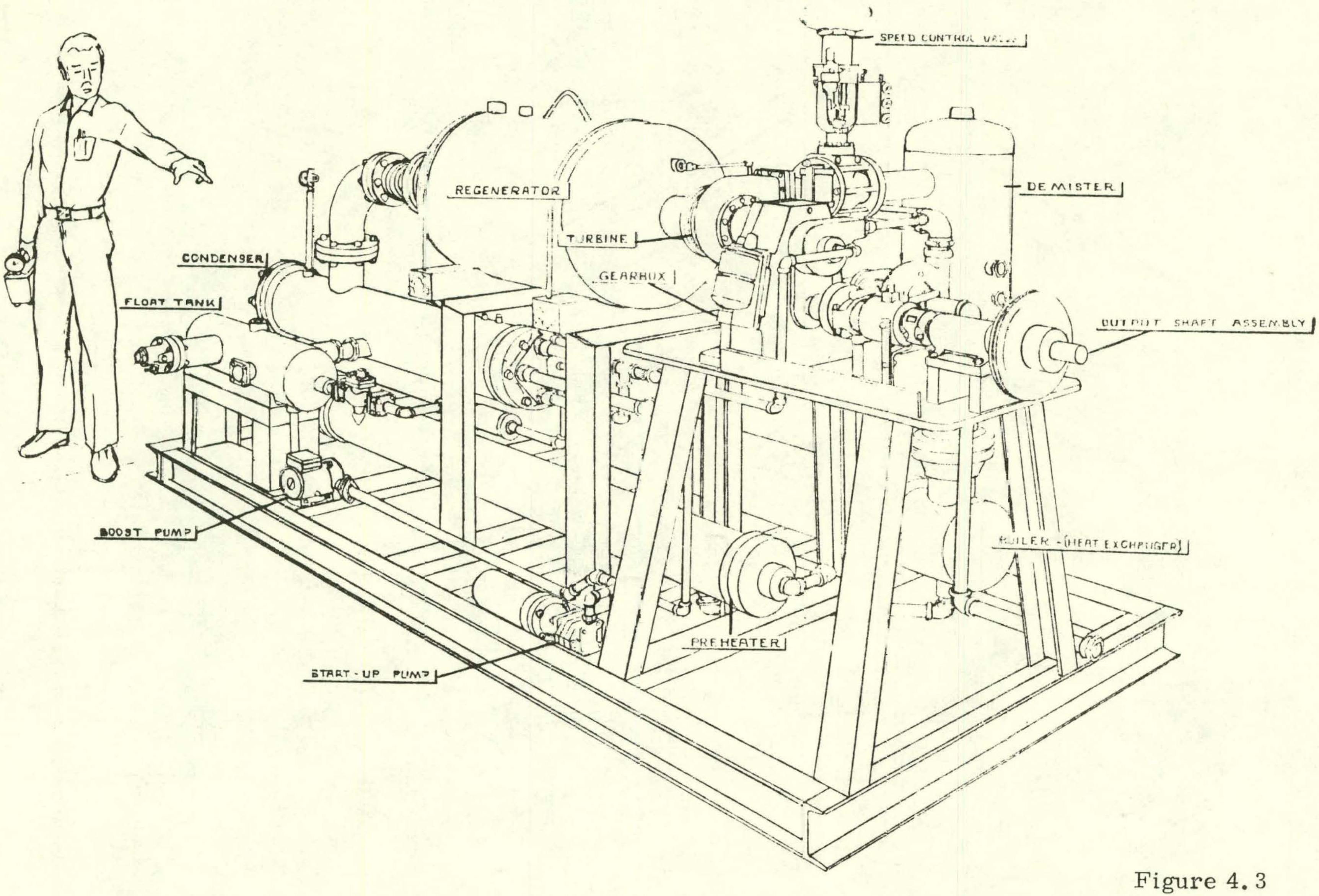


Figure 4.3
25 HP RANKINE ENGINE
BARBER-NICHOLS
ENGINEERING CO.

store heat, both for the short term, passing cloud, and for operation into the after-sundown time period. The alternative to this heat storage that is used in the BIC system is to make a larger engine and store the power produced, either as pumped water or electricity pumped into a grid.

The concept of boiling in the collector was first considered by Barber-Nichols in 1976. Some laboratory experiments were conducted in early 1977 using ten-foot sections of electrically heated tubes. These tests simulated the heat flux and the unilateral heating pattern of solar receiver tube. These tests were successful and demonstrated the feasibility of the concept. They also gave some guidance on the required geometry of the system and a means of controlling the fluid levels.

This report describes the results of carrying these feasibility studies to full size collectors using solar energy as the heating medium. Thus, actual field conditions were tested, including the associated problems of building a leak-free system, maintenance, etc.

5.0 COMPARATIVE COST ANALYSIS

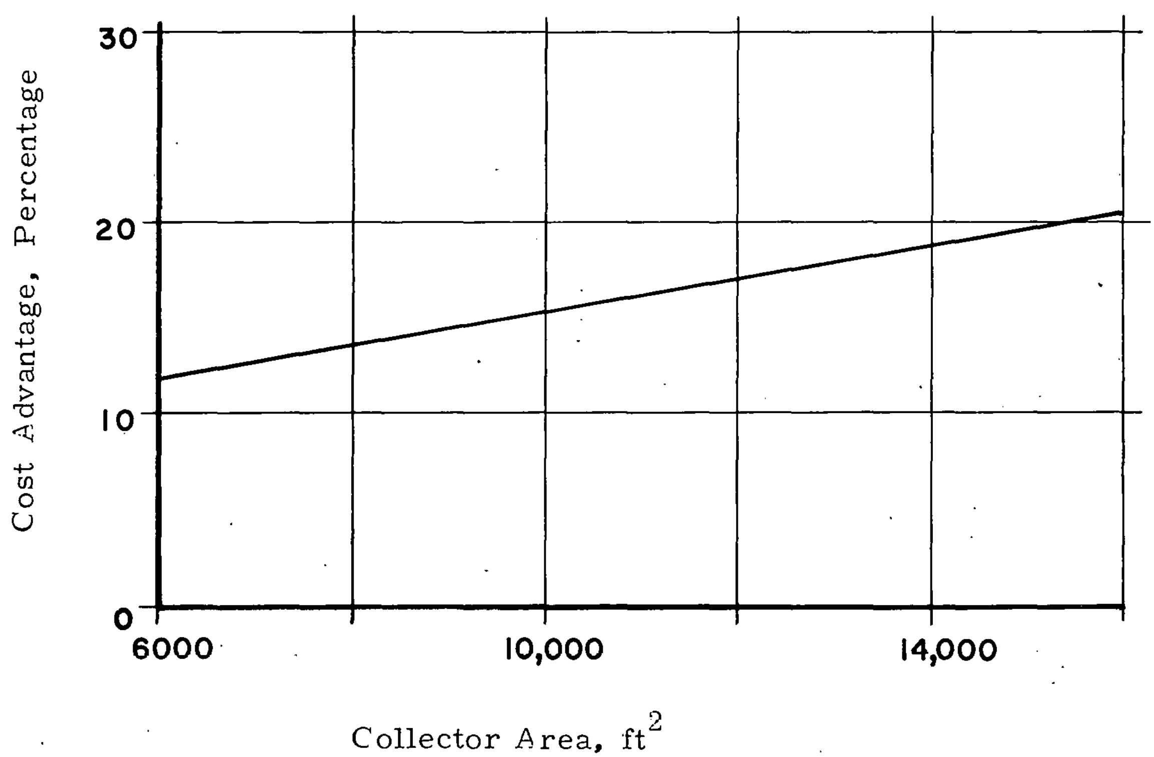
The cost advantage of the boil-in-collector system over the conventional system can be demonstrated, given some basic assumptions. For a specific collector field size, the two systems require quite a different size of heat storage and engine. Therefore, the result is closely tied to the assumption of costs for these components. The assumed costs were supplied by Sandia based on costs at Willard (2). The detailed analysis is carried out in Appendix A.

The important result is that the boil-in-collector system has a cost advantage of 10 to 20 percent (Figure 5.1) with the cost advantage increasing with larger collector systems that can either operate longer hours in the case of the conventional system with oil storage system or greater engine power for the boil-in-collector system (Figure 5.2). Basically, given a collector system size, the cost of the storage system (tank, oil, pump, etc.) and the extra heat exchanger for the conventional system is more than the cost of larger engine (without the heat exchanger) for the boil-in-collector system (Figure 5.3). Essentially, the scheme is to store water rather than heat.

For systems in the megawatt range, there would be added costs in the boil in collector system to build a framework to support the collectors at the 5 degree angle unless there was a suitable natural slope on which the collectors would be placed. Also, for the megawatt sized system, there could be some cost increase for the large, high pressure vapor piping. However, for the present sized system (tens of horsepower) which would generally be used for a distributed system, the boil-in-collector system is more economical.

FIGURE 5.1

Cost advantage of boiling in the collector as compared to the conventional oil system.



$$\text{Cost Advantage} = \frac{\text{Cost of boil-in-collector system}}{\text{Cost of conventional system}}$$

FIGURE 5.2

Comparison of the operating hours for the conventional system with the engine size for the boil-in-collector system as a function of collector area assuming constant engine efficiency.

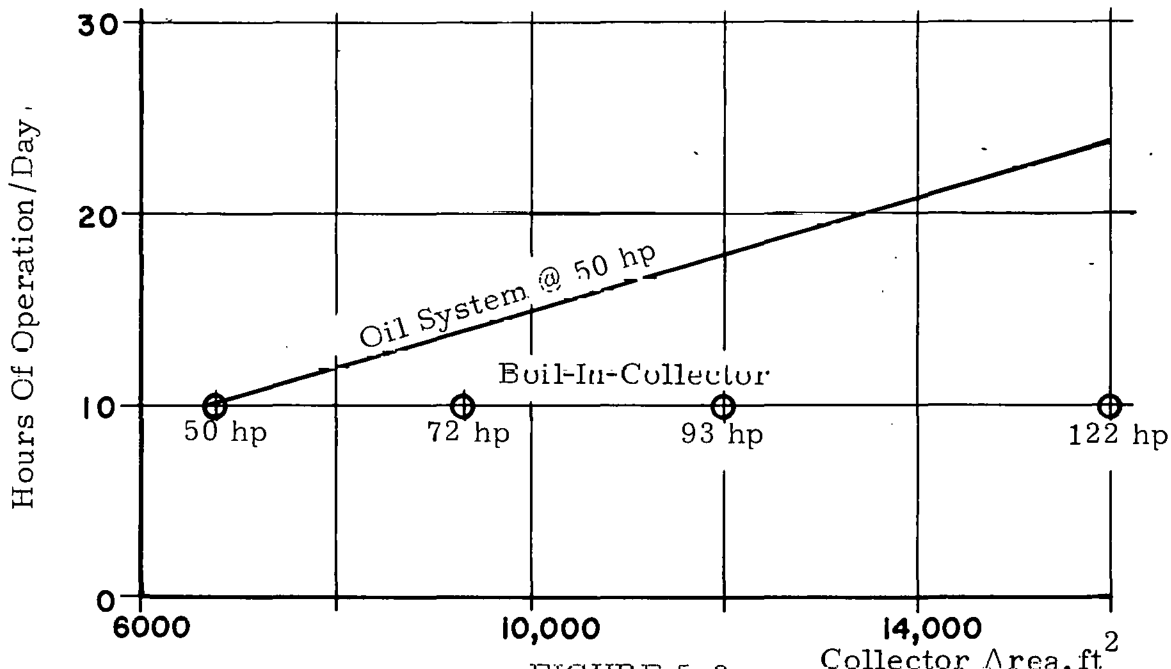
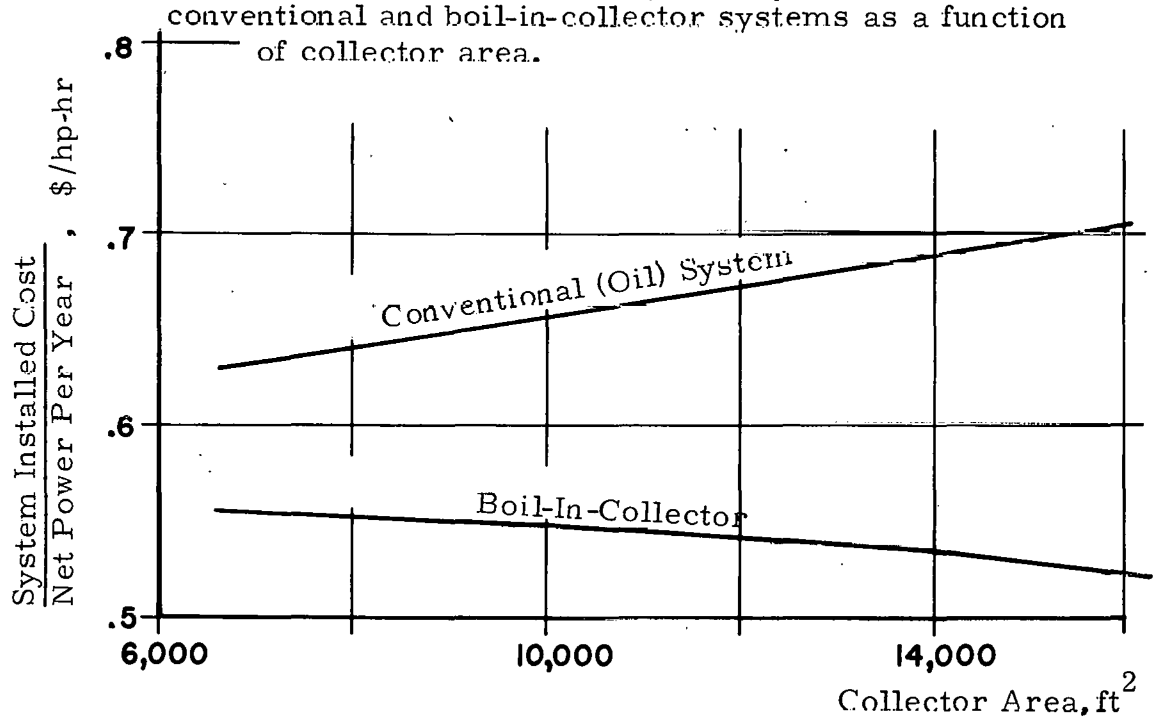


FIGURE 5.3

Comparison of installed cost per net power out for the conventional and boil-in-collector systems as a function of collector area.

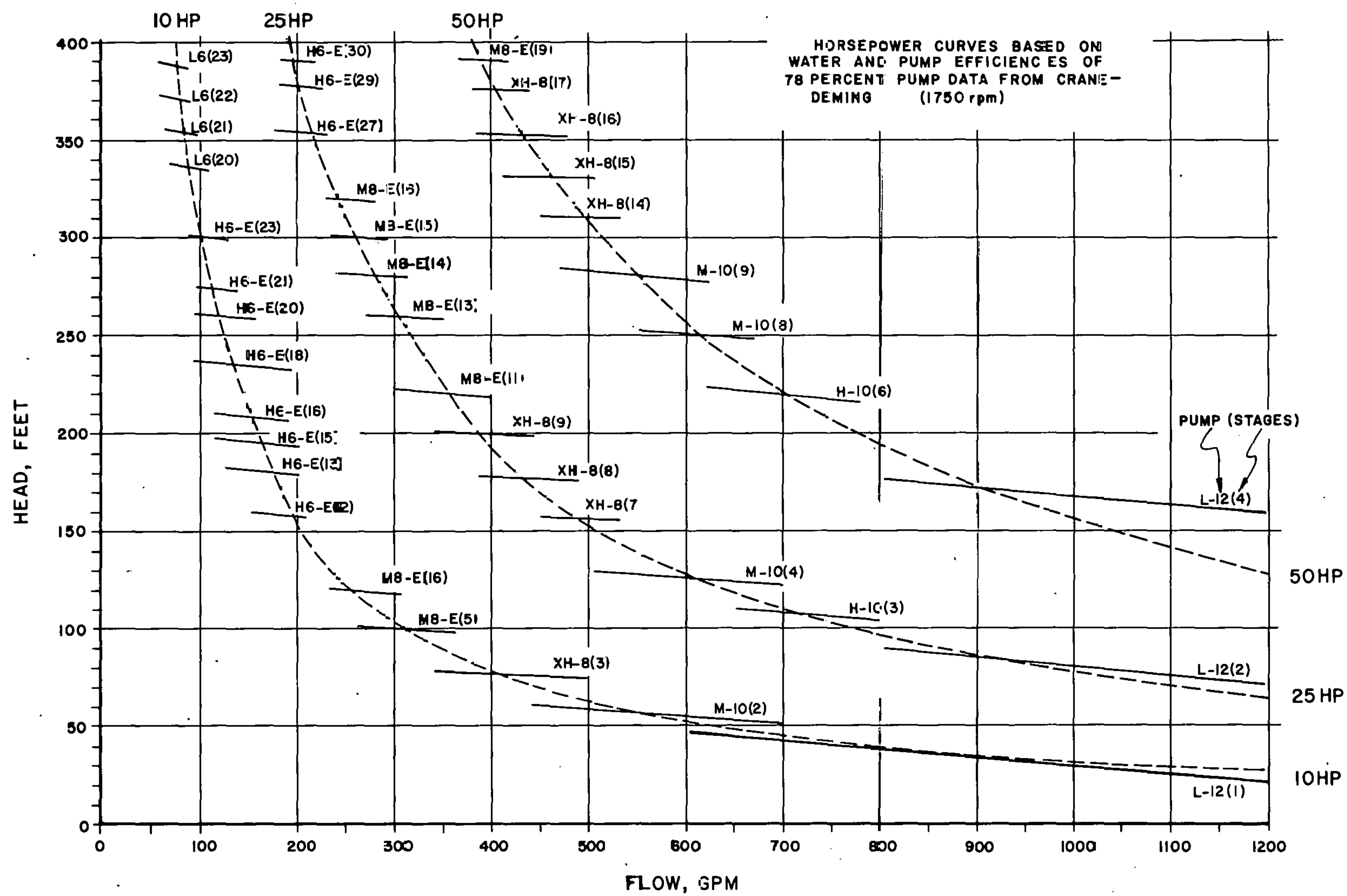


6.0 PUMP MATCHING AND MARKET SURVEY

A survey of the various pump sizes presently used along with the well depths has been conducted, and is presented in detail in Appendix B. This, along with the type of water distribution system (sprinkler or ditch) would give the engine size, head, and flow required. As presented in the detailed report (Appendix B), well depths are highly variable and much dependent on local geology. Given a certain location and prescribed well depth, the pumping rate is dependent on the flow capability of the well (draw-down characteristics) and on the crop requirement. It appears that the general situation is to provide an engine sufficient to pump the well at whatever rate it is capable of flowing, rather than size it for the crop.

The engine supplied by Barber-Nichols for this test has heat exchangers and rotating machinery capable of producing 25 horsepower. However, as this would require a large (over 2000 ft^2) collector field, the turbine flow rate has been scaled down to about 2 horsepower. Looking at Figure 6.1, (extracted from the detailed report in Appendix B), it can be seen that this engine, either as a full 25 horsepower, in a reduced form (10 horsepower), or as a doubled (50 hp) unit, is capable of covering a large range of head-flow. The short lines crossing the horsepower lines delineate pump sizes and numbers of stages from a particular pump manufacturer. Thus, there are currently available pumps that will match the engine for virtually any desired combination of head-flow.

FIGURE 6.1
IRRIGATION PUMP CHARACTERISTICS



7.0 SYSTEM DESCRIPTION

The system consists of two major subsystems: the solar collector subsystem and the Rankine engine subsystem (Figure 7.1). In the present test facility the Rankine engine subsystem does not have a turbine-gearbox installed. The turbine is simulated thermodynamically as the R-113 is throttled through a valve to the condenser. The Rankine engine subsystem acts as a heat sink for the collector system, and as a source of high pressure, cold liquid for collector feed. It also provides some control functions.

The collector system consists of one row (160 ft^2) of Solar Kinetics (Dallas, TX) collectors and two rows (128 ft^2) of Del Manufacturing (Los Angeles) collectors. These three rows of collectors are mounted on a space frame installed on the roof of Barber-Nichols Engineering Company in Arvada (a northwest suburb of Denver) (Figure 7.2). The collectors are oriented with their axis North-South and they track East-West. They were inclined 5 degrees up from the horizontal (North end elevated). This was to provide for the natural circulation of the boiled refrigerant up and out of the high end of the tube with cold liquid in at the bottom end.

The space frame was constructed in the form of a large platform (20 feet wide by 40 feet long) pivoted on the South end with adjustable support columns on the North end. This frame allows for various inclination angles from 0° to 10° above horizontal. The platform was constructed from three 40-foot long bar joists (roof trusses) 24 inches high. These three trusses served as the rail on which each of the three rows of collectors were mounted. The collector pylons were mounted on plates attached to the trusses. The trusses had suitable cross braces to eliminate sway and deflections due to wind loads or weight. The entire space frame was supported on the walls and on the existing steel framework of the building.

The collector selection process involved contacting a number of manufacturers for their interest, analyzing the response to determine if the collector was suitable for the boiling-in-collector concept, and determining if the efficiency was suitable. This is discussed in a detailed report in Appendix C. In comparing the various collectors systems the more important considerations were 1) suitability for the natural circulation boiling-in-collector concept, 2) receiver tube heat flux (concentration ratio), 3) efficiency, 4) quality of construction, and 5) method of handling the rotation of the receiver tube. The last consideration is very important for the boiling concept as the fluid is necessarily high pressure in addition to high temperatures. Also as it is necessary to have a leak-free system to prevent loss of the working fluid, the means of obtaining rotation of the collector without slip joints is important.

As described in Appendix C, the two collectors selected were manufactured by Solar Kinetics and Del Manufacturing Company. These have some

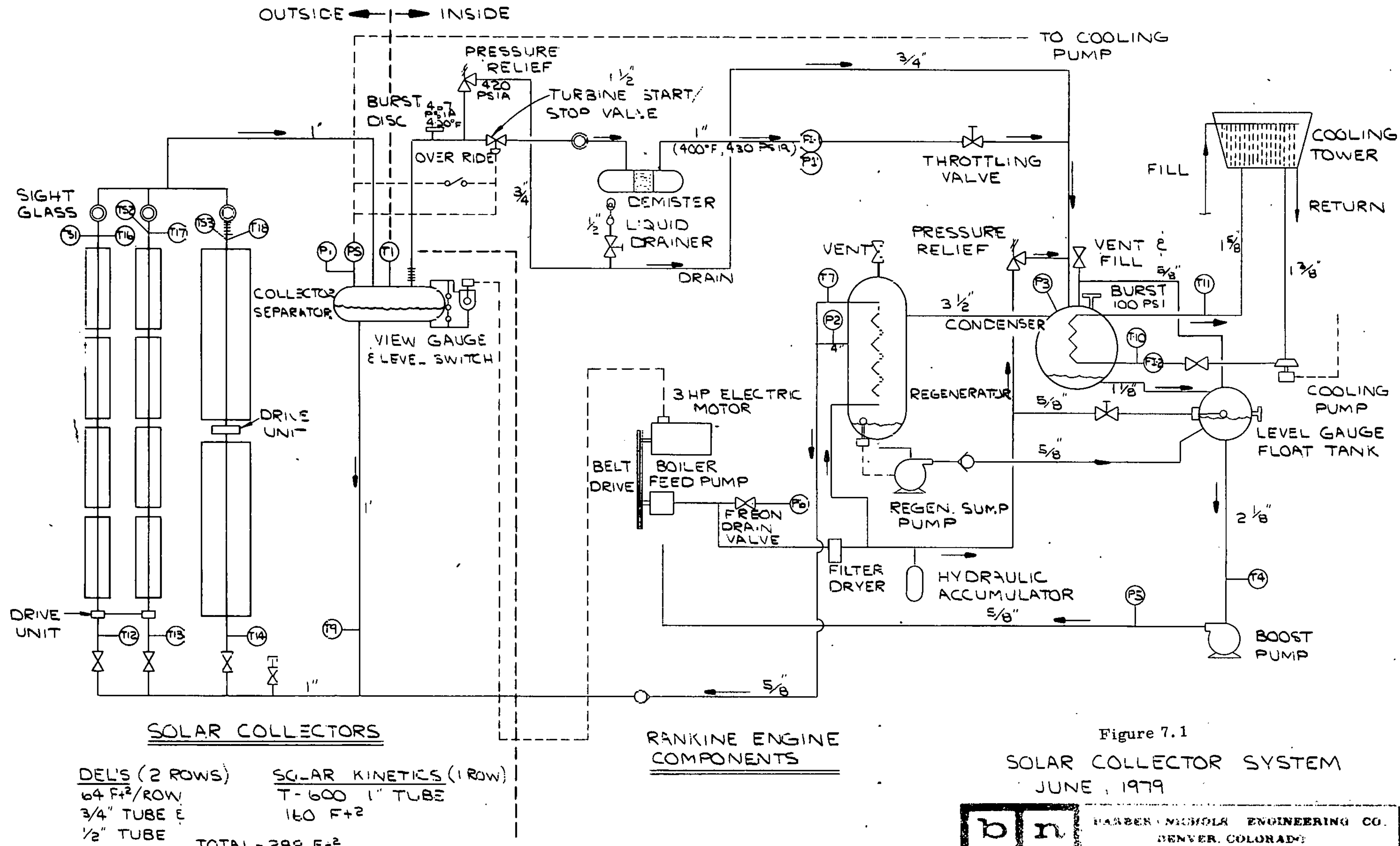


Figure 7.1

SOLAR COLLECTOR SYSTEM

JUNE, 1979

BARBER-NICHOLS ENGINEERING CO.
DENVER, COLORADO

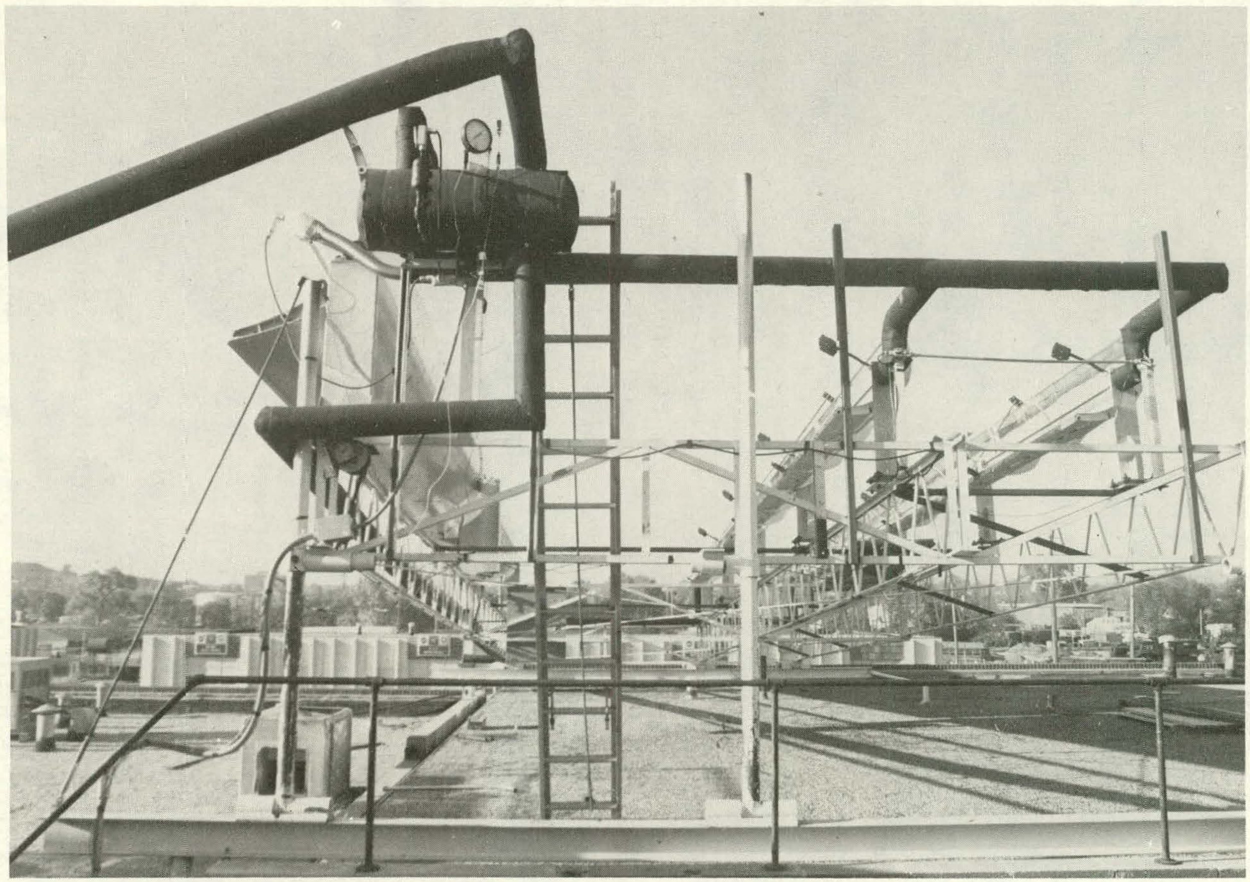


Figure 7.2 Overall View of Collector System Looking South

distinct differences that made it worthwhile to test both types. The Solar Kinetics have a larger concentration ratio (38:1 compared with 32:1 for Del). The Del unit rotates about a fixed receiver tube allowing a very desirable means of connecting the collector to the header without rotating joints, flexible tubes, etc. Rotating about the receiver tube is unique to the Del collector and the hard plumbing is very useful to the boiling-in-collector concept. The Solar Kinetics collector pivot point is about one foot from the receiver tube focal line, thus requiring the use of a flexible tube or rotary unions. It was felt that a collector with this type of flexible joint was typical and therefore should be tested. Both the Del and Solar Kinetics collectors have approximately the same projected efficiencies.

The collectors chosen are slightly different than those advertised by the two firms. The Solar Kinetics collector has air in the annulus around the receiver tube rather than a soft vacuum or Argon. Also, the receiver tube is 1.3 inches in diameter rather than 1 inch. The Del collector has a 3/4 inch diameter receiver rather than a half inch receiver. Both of these changes cause the efficiency to be slightly lower than advertised. The two collectors selected have a combined system (Rankine engine - collectors) efficiency peaking at a temperature slightly above 400^{o}F for a high (300 Btu/hr-ft^{2}) solar flux and a peak efficiency occurring between 300^{o}F and 400^{o}F for a more moderate flux (200 Btu/hr ft^{2}) (Figure 7.3). Therefore, the system nominal design point was chosen to be 350^{o}F.

The Solar Kinetics collectors are shown in Figure 7.4. They are hydraulically driven with the center pylon being the drive unit. The tracking system is the Delavan Sun-Lok I. Barber-Nichols provided an over-temperature control and a stow switch. Mercury switches in the tracker provide limit stops. The receiver tube is 1 1/4 inch diameter steel with black chrome oxide coating. This is covered with a pyrex glass envelope. The reflector is an aluminized acrylic film attached to the aluminum monocoque structure. The module is 4 feet wide (aperture) by 20 feet long. The receiver tube is attached to a flexible metal hose at each end of the 2-module row.

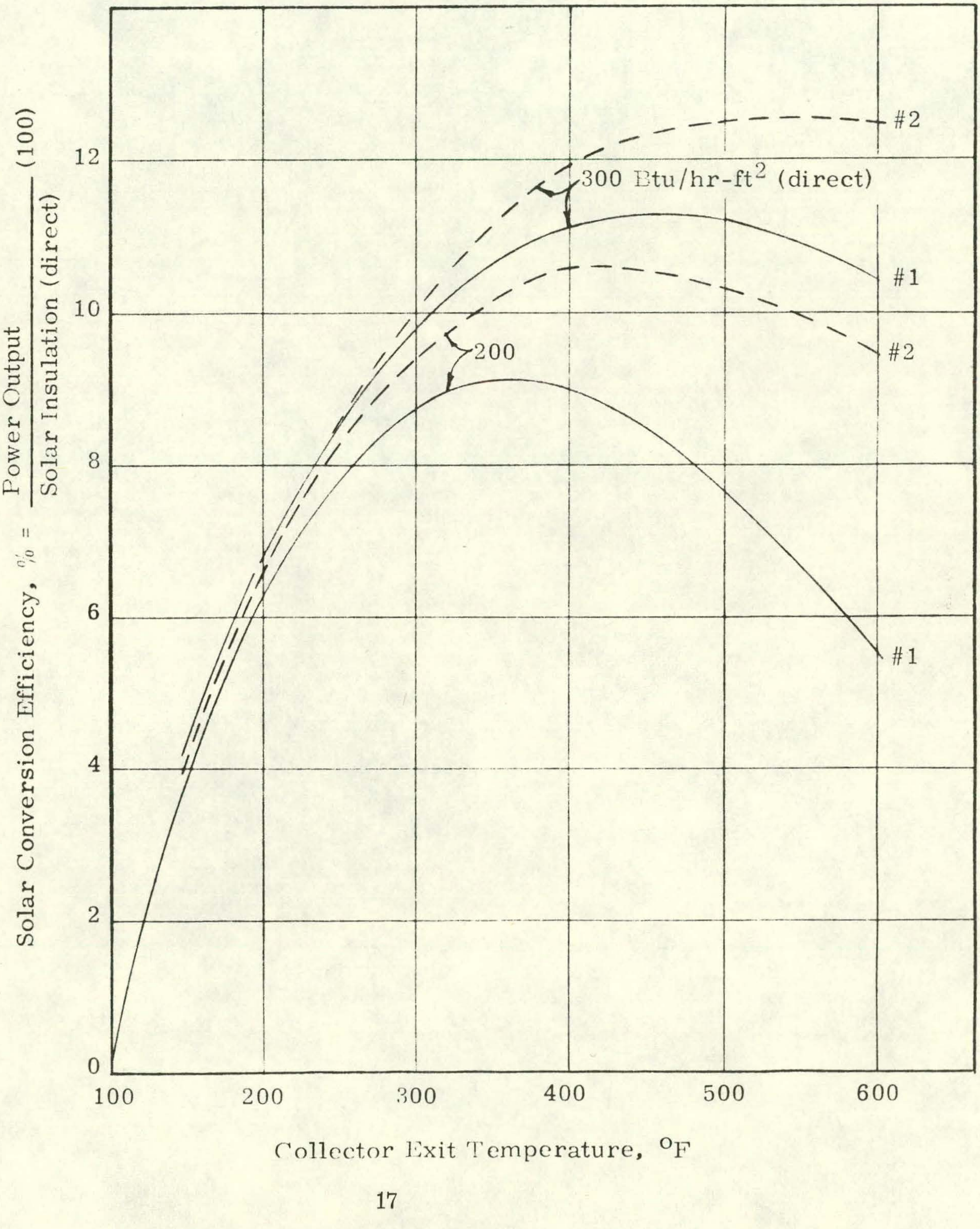
The Del collectors are shown in Figure 7.5. They use sagged glass-silvered backed mirrors supported in an open framework. The modules are 2 feet wide (aperture) by 8 feet long. The drive is an electric motor-driven worm-drive gearbox at the lower end of the row. A drive shaft connects the gearboxes of the two rows. The tracker is a Delavan Sun Lok I. The receiver is 0.75 inches in diameter, (necks down to 1/2 inch at the ends) and is steel. The surface is coated with black chrome. A pyrex glass tube (air filled) covers the receiver with the air in the annulus passing through a dessicant when entering the annulus.

The collectors are connected by headers along the top and the bottom of the rows (Figure 7.6). The supply line from the engine feed pump enters the lower (liquid) header at the east end which happens to be the Solar Kinetics end. The upper (vapor) header has a riser that connects to the receiver tank. This tank acts as a liquid separator with a liquid return line to the

FIGURE 7.3

ESTIMATED SOLAR CONVERSION EFFICIENCY

25 Horsepower



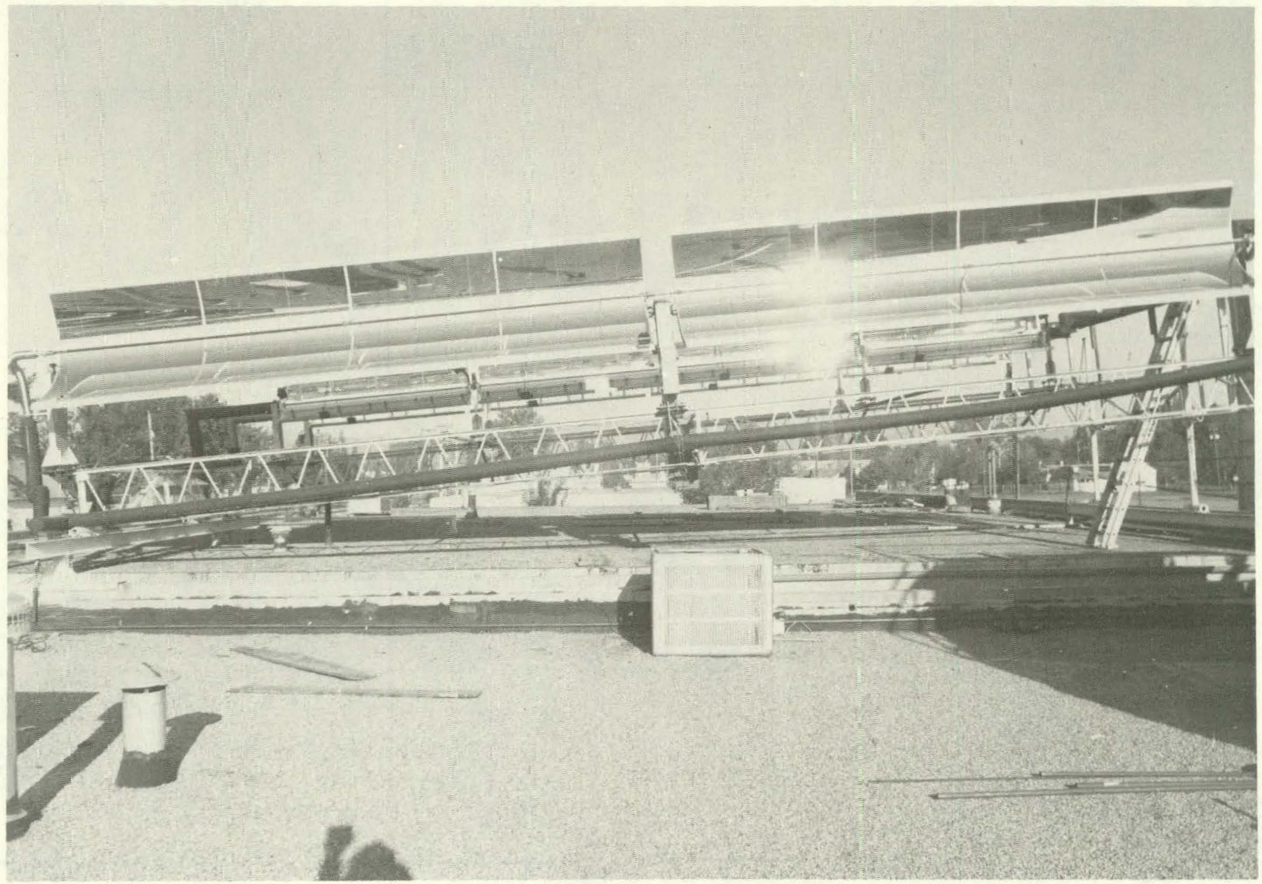


Figure 7.4 Solar Kinetics Collectors Viewed Frcm the East

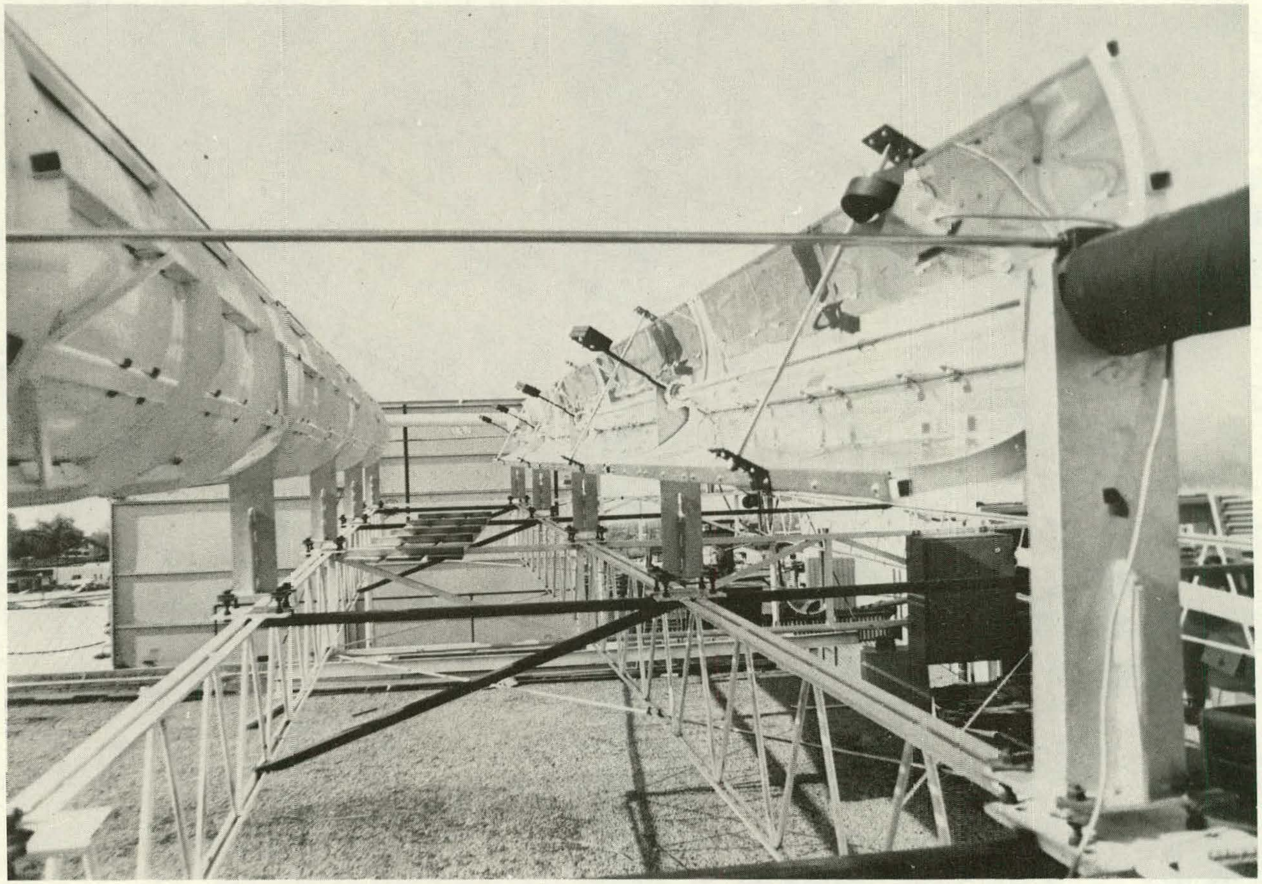
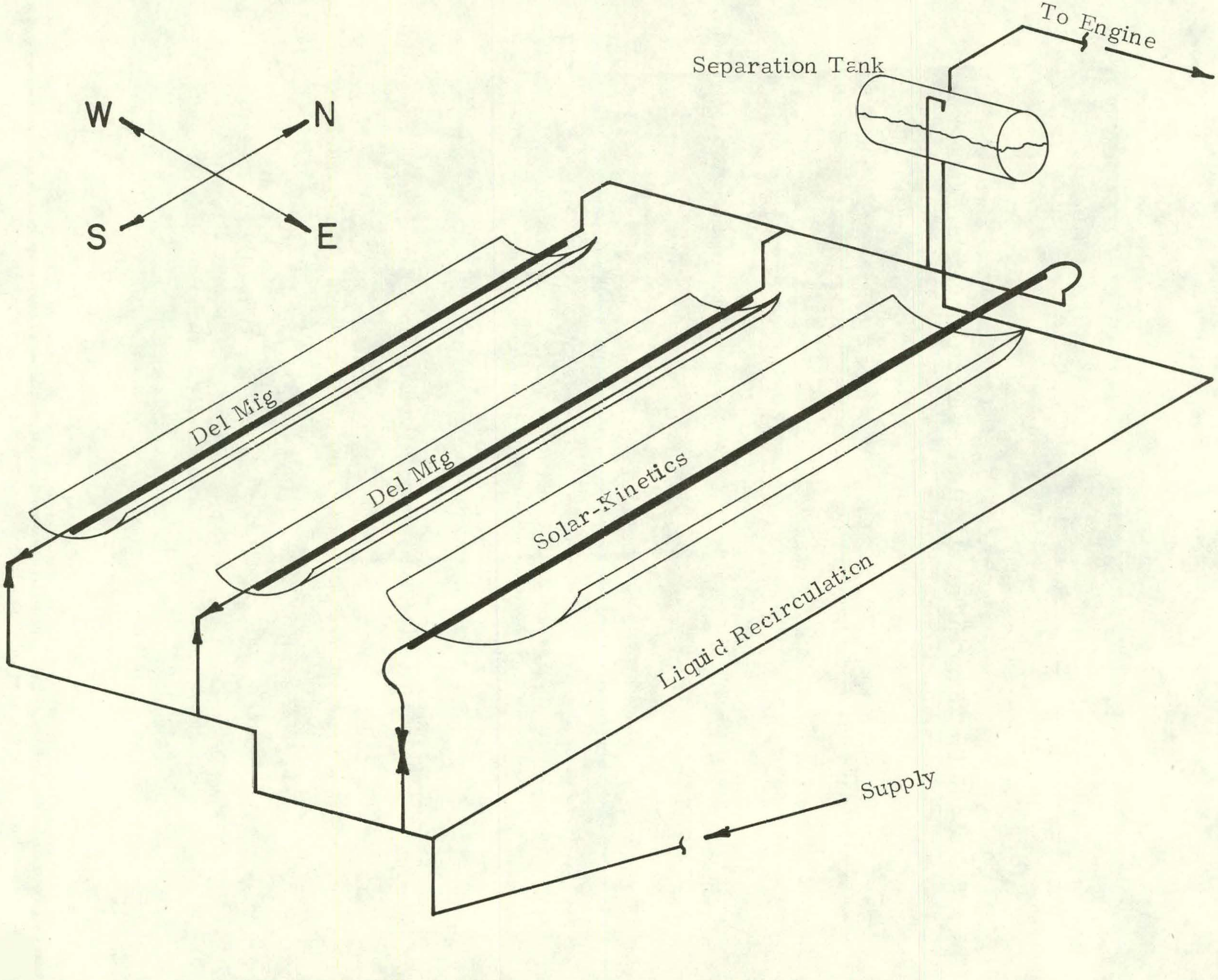


Figure 7.5 Del Manufacturing Collectors Viewed From the North

FIGURE 7.6

ISOMETRIC COLLECTOR PLUMBING



lower (liquid) header. There is also a vapor line leaving the top of the receiver tank going to the Rankine engine. The receiver tank has a liquid level gage and a liquid level switch. This switch turns the feed pump on and off to maintain the receiver tank half full.

The Rankine engine subsystem includes a condenser, regenerator (not necessary for the system as used), a feed pump, motor controls, and other ancillary equipment (See Figures 7.7 and 7.8). The gearbox is complete but the turbine wheel casting and housings have not been machined. Therefore, for the collector tests, a valve replaced the turbine. This Rankine engine is similar to other Rankine engines except that its physical size is quite small for its rating (25 hp). Figure 7.9 shows the 25 hp engine at Willard which is much larger and more complex than the engine of Figure 7.7 and 7.8. This reduction is due to the boiling-in-collector concept and its lack of requirements for heat exchangers. Two views of the engine installation are shown in Figures 7.7 and 7.8. The engine was installed on the balcony of a building adjacent to the collectors. Thus, the engine was on the same level as the base for the collector system.

The turbine would be an axial flow turbine running at a speed of 19,500 rpm. This type of wheel has the capability of using one or any number of nozzles to provide various levels of power output. There is only a slight performance penalty due to the use of less than a full compliment of nozzles. The gearbox would slow the turbine speed down to 1750 rpm. The turbine was not completed for this test.

The feed pump is a Dynesco, diaphragm type pump driven by an electric motor. When the turbine is installed, the gearbox output would drive the feed pump directly, making use of an electric clutch to control the liquid level in the collector receiver tank.

The complete system schematic was shown in Figure 7.1. Sufficient instrumentation (as shown in Figure 7.1) was provided to control the system and determine information of interest. Also, safety devices, control switches and valves, start up and shutdown devices, and other ancillary equipment were provided.

The heat from the collectors was throttled across an adjustable hand valve. The high energy R-113 then entered the condenser and was condensed to a liquid, ready to be pumped back into the system. The heat rejected in the condenser was removed by cooling tower water.

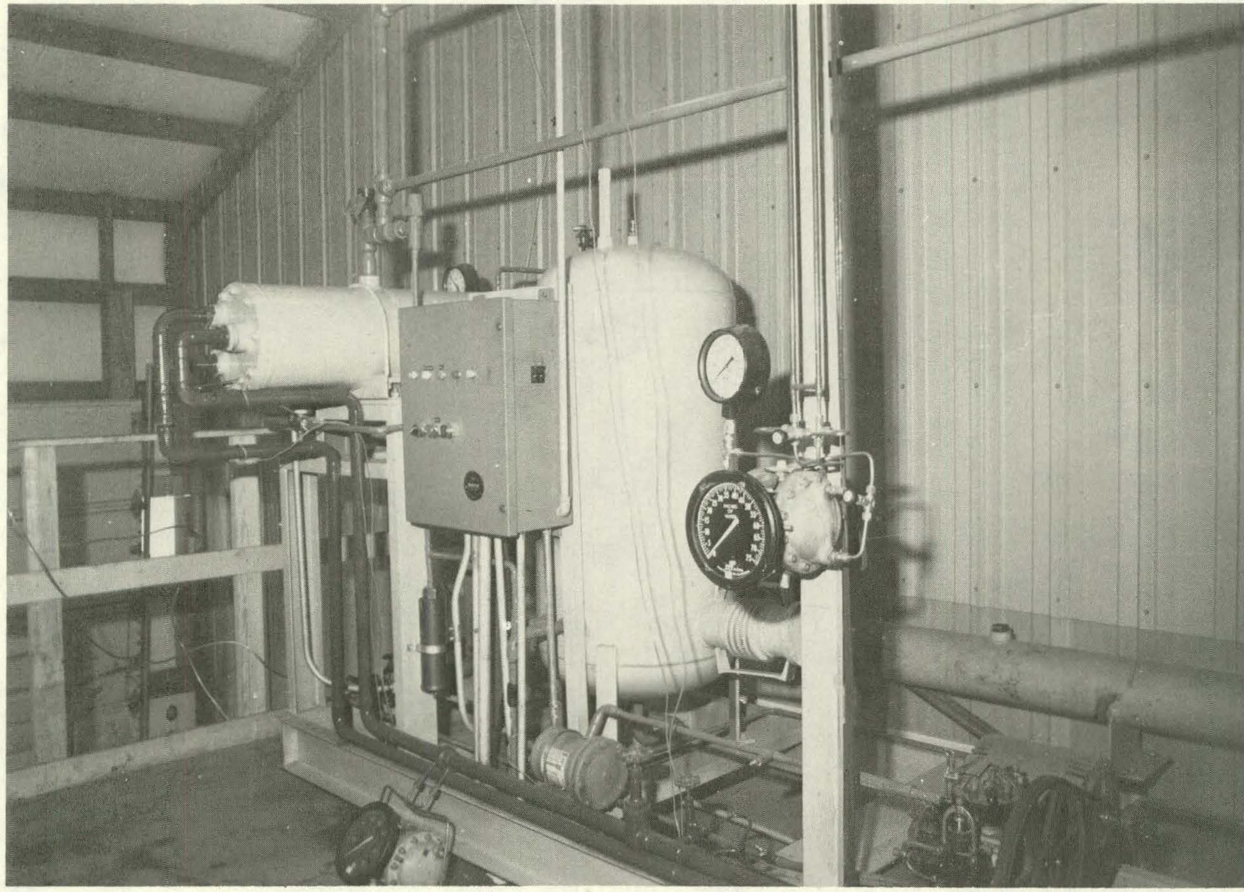


Figure 7.7 Rankine Engine Viewed From Turbine Mounting End

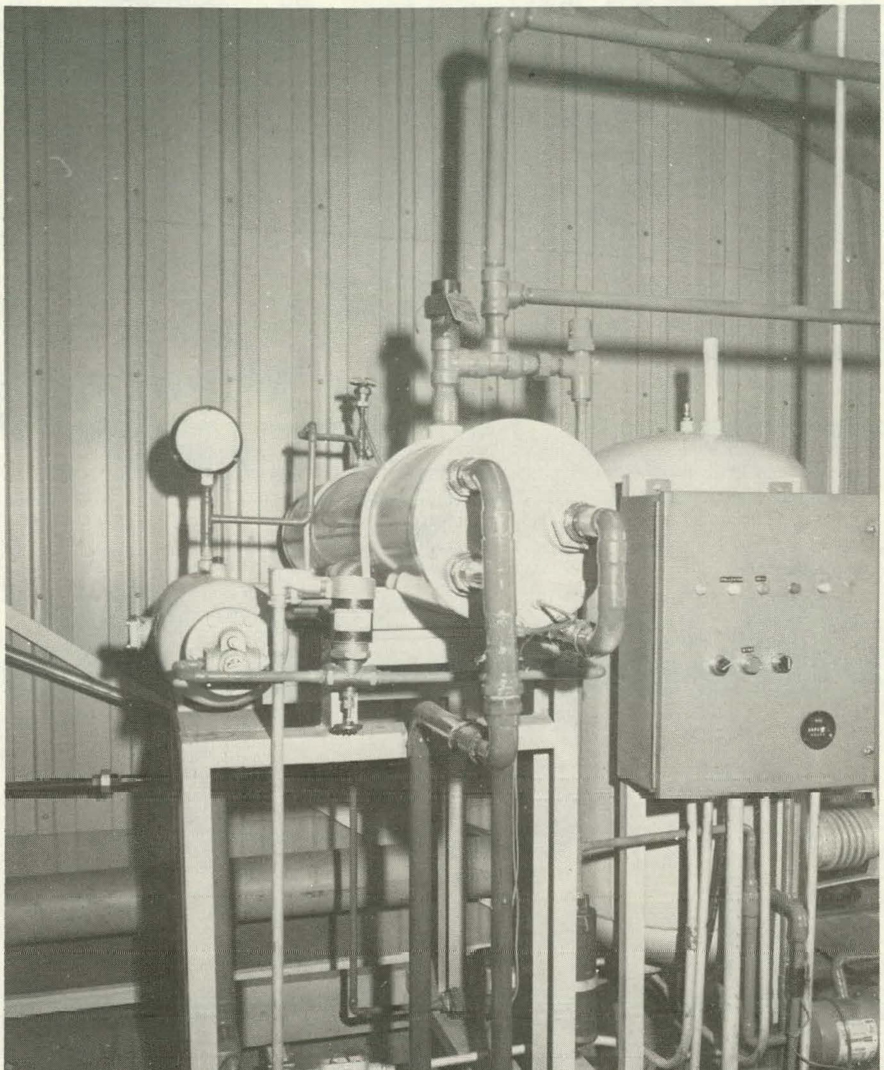


Figure 7.8 Rankine Engine Viewed From Condenser End

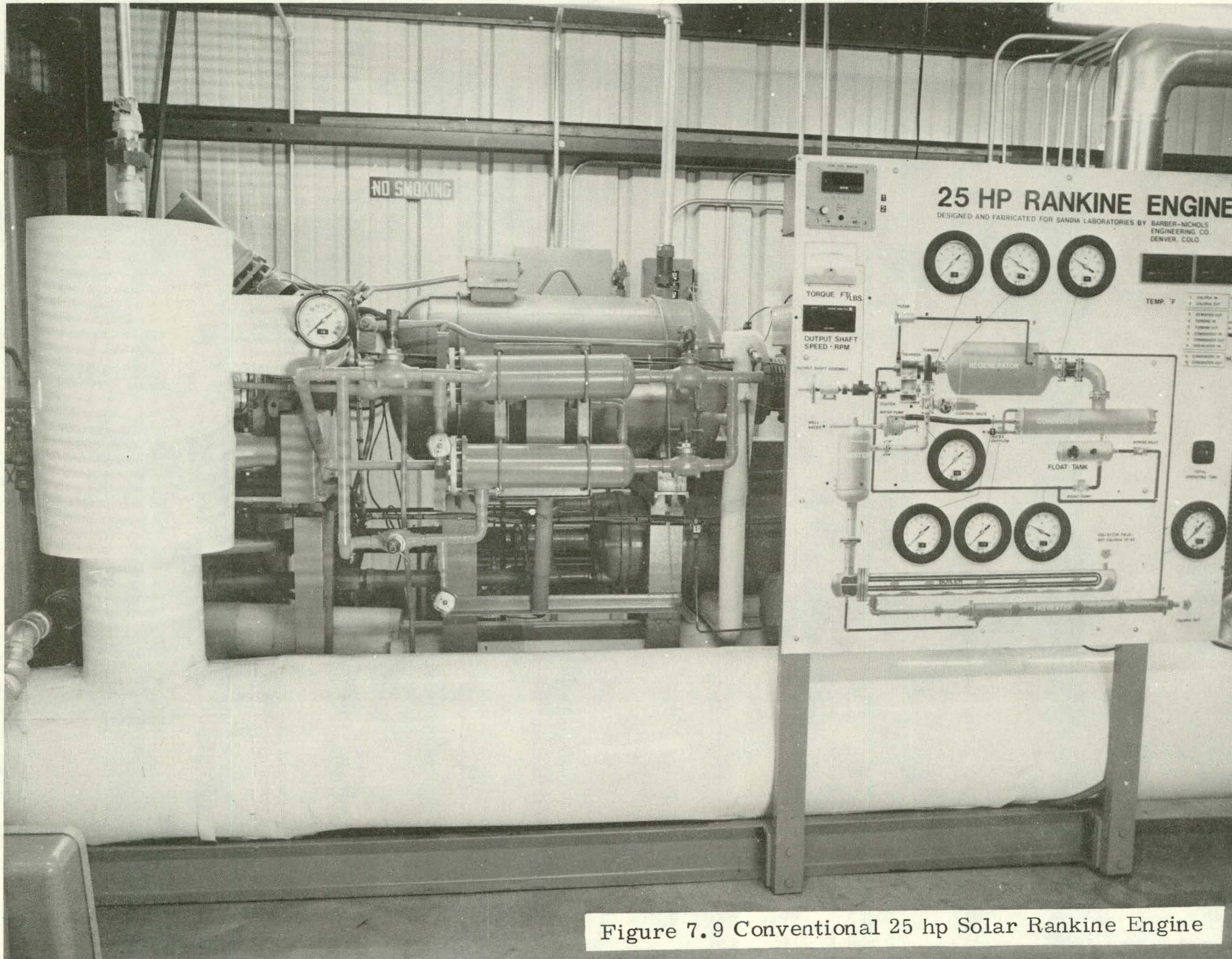


Figure 7.9 Conventional 25 hp Solar Rankine Engine

8.0 METHOD OF ANALYSIS

The equations for calculating the test quantities desired are shown in Table 8.1. Each of these will be discussed in turn with particular attention given to how each parameter was measured or determined. This, in turn, will be incorporated in an error analysis which will give an estimate of the error involved in the results.

The solar energy available to the collector is the product of the solar flux incident on the collector and the aperture area (eq. 1, table 8.1). The collector aperture area is simply the product of the aperture width and the collector length. The solar flux is somewhat more complex. Since the parabolic collector can only make use of direct (beam) radiation, the measurement must be the direct radiation normal to the sun with a correction for the end effect. These end effect connections were not made since the data was collected in June (high sun angles) and the collectors were tilted up 5 degrees. Solar Kinetics verified (on their computer) that the end effect correction was negligible.

The measurement of direct (excluding diffuse) normal (to the sun) radiation is difficult and requires an expensive pyroheliometer. A hand held total (hemispherical) radiation meter (Mark VI Sol-A-Meter, Matrix, Inc., Mesa, Arizona) was used. This device was hand oriented to be normal to the sun. These readings were compared with direct normal readings made at NOAA laboratories in Boulder during the same time frame. The discussion of this comparison will be made in Section 9.0.

The energy collected (equation 2, table 8.1) is the product of the refrigerant mass flow and the specific enthalpy increase in the refrigerant. The mass flow was determined using an ASME sharp edged orifice. This was installed in the vapor line just before the throttle valve into the condenser. The enthalpy out of the collector system was determined by the pressure and temperature at the inlet to the throttle valve. The temperature was measured using a thermocouple. The enthalpy into the collector system was determined from the liquid temperature as measured at the regenerator outlet using a thermocouple (liquid).

The energy delivered to the condenser was determined on the water side of the tube-in-shell condenser using the water flow and the temperature rise of the water (eq. 3, table 8.1). The water flow measurement was made with an ASME code sharp-edged orifice. The temperatures were measured using thermocouples. Flow rates were controlled so the temperature difference of the water exceeded 10^{o}F.

The heat leak in the collector system was computed by using a standard heat transfer analysis assuming the pipe temperature was known and that the wind velocity was 7.5 mph (eq. 4). The ambient temperature

TABLE 8.1

CALCULATION PROCEDURE

SOLAR ENERGY AVAILABLE TO COLLECTOR: (Eq. 1)

Q in = (Solar Flux) * (Collector Aperture Area)

Where the Solar Flux is the Total Normal To Sun - measured with a Mark VI Sol-a-Meter, Metrix, Inc., Mesa, Arizona

ENERGY COLLECTED: (Eq. 2)

Q collected = (Mass Flow R-113) * (Enthalpy @ Throttle valve in - Enthalpy @ regenerator out)

ENERGY INTO CONDENSER: (Eq. 3)

Q condenser = (Mass Flow Water) * (Cp) * (Temp out - Temp in)

HEAT LEAK IN SYSTEM: (Eq. 4)

Qloss = Heat loss based on heat transfer calculation assuming a 7 1/2 mph wind.

COLLECTOR EFFICIENCY: (Eq. 5)

$$\eta_c = \frac{Q\text{ collected} + Q\text{ loss}}{Q\text{ in}} * 100$$

CONDENSER HEAT BALANCE: (Eq. 6)

$$H.B. = \frac{Q\text{ collected} - Q\text{ condenser}}{Q\text{ collected}} * 100$$

was measured using a thermocouple. The pipe temperature was taken to be the saturated liquid temperature as measured by the tank temperature.

The collector efficiency is computed as the heat picked up by the collector (eq. 2) plus the heat lost (eq. 4) divided by the heat incident in the collector (eq. 1). This is shown as equation 5.

The heat balance is a measure of how well the heat delivered by the system (eq. 2) is measured relative to the heat picked up by the condenser (eq. 3). This is depicted in equation 6 where a positive result would indicate less heat delivered to the condenser water than picked up by the collector.

9.0 PERFORMANCE AND ANALYSIS

The collector system was operated during the months of May and June, 1979 on an irregular basis. Hardware problems (the tracker) and unsuitable weather caused the spasmatic operating schedule. The Del collectors were ready for operation first as one of the Solar Kinetics collectors was damaged in shipment and had to be replaced. It was quickly determined that the sagged glass mirrors in the Del collectors were of such poor quality that extensive testing was not justified. Thereafter, most testing was done with the Solar Kinetics or the combined field of Del and Solar Kinetics.

The collector efficiency as computed from the various tests are shown in Figure 9.1. This plot does not include all the data taken as only representative points were reduced from a particular days operation. A plot of a typical days data is shown in Figure 9.2. This set of data is for the Solar Kinetics collector. An example of the reduction of a given set of data is shown in Table 9.1, which is an output sheet from a computer program.

The daily run data in Figure 9.2 shows that the system operates in a steady manner with only minor variation. At approximately 13:30 some high, wispy clouds caused a small decrease in the solar flux which was also picked up in the other parameters. The system pressure was manually controlled to maintain a nearly constant collector out temperature (system pressure constant) so the decrease in solar flux at 13:30 caused a drop in mass flow but not in temperature. The tank temperature shown on the figure is the temperature of the receiver tank that separates the liquid from the vapor at the collector exit. As this tank operates in a saturated condition, the temperature is a function of the system (tank) pressure. The system out temperature was measured at the throttle valve which replaces the turbine for this test. This temperature is nearly equal to the tank temperature, as it should be. The collector outlet temperature is used only for control. In this particular set of data the collector outlet temperature measured somewhat below the tank temperature, which could not be the case. The thermocouple for this measurement was moved to a different location at a later date which then corrected this problem. The temperature into the collector is a measure of the mixed stream coming from the engine (system temperature in) and the recirculation loop liquid. The low temperature here indicated that only a small amount of recirculation is occurring.

The testing was hampered by the poor performance of the Delavan Sun Lok I tracking unit. The unit on the Solar Kinetic collector worked well when received. The unit on the Del collector would track a bright sun, but when the slightest decrease in solar flux took place, the unit would track away from the sun. This was diagnosed as differences in characteristics in the phototransistors. The unit was rebuilt using different phototransistors with some changes in circuiting to avoid this problem. The tracker on the Solar Kinetics unit began to experience the same problem late in the test

FIGURE 9.1

Test Results 5° Inclination

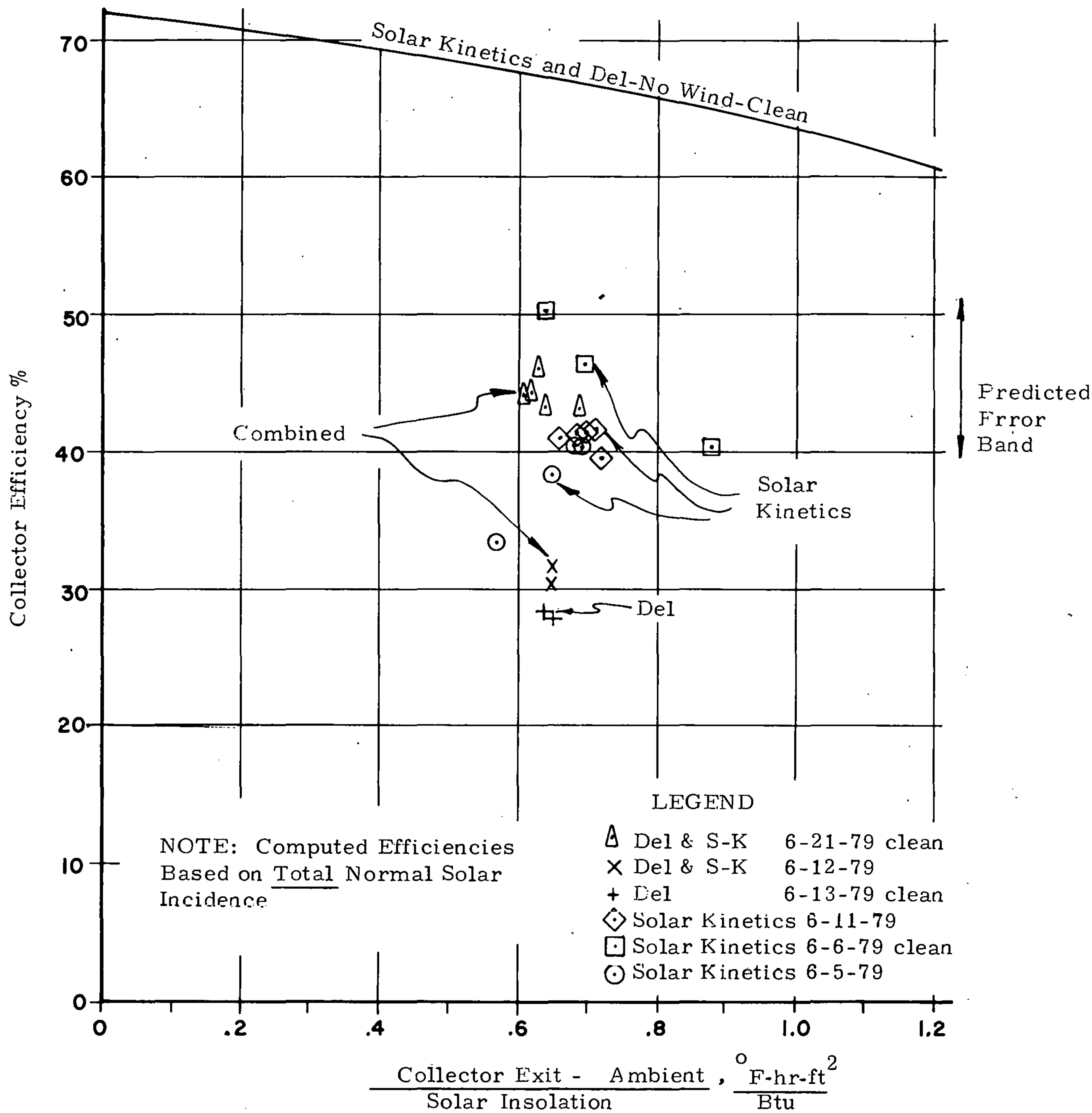
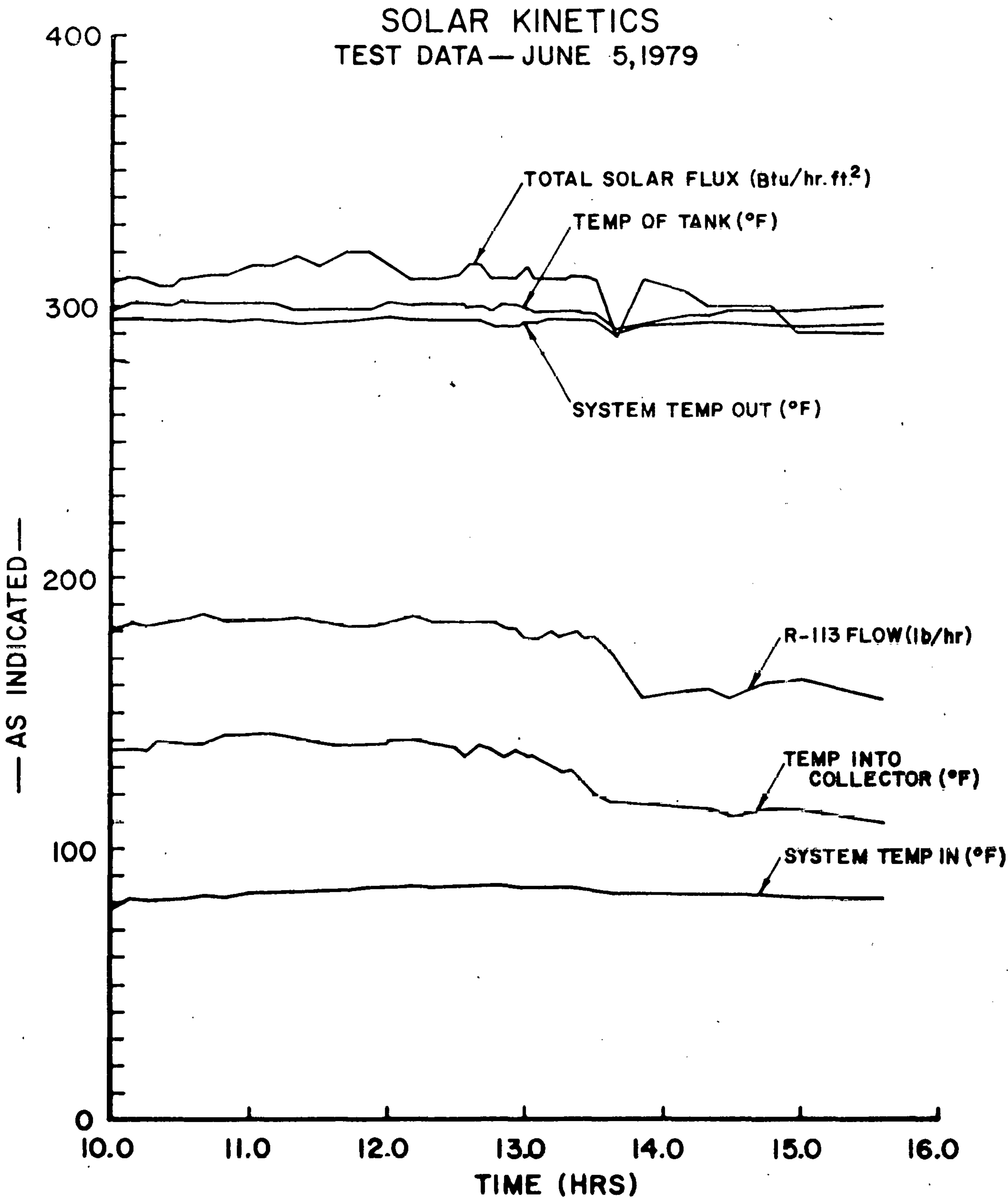


Figure 9.2


SOLAR KINETICS
TEST DATA — JUNE 5, 1979
TOTAL SOLAR FLUX (Btu/hr.ft.²)
TEMP OF TANK (°F)
SYSTEM TEMP OUT (°F)
R-113 FLOW (lb/hr)
TEMP INTO COLLECTOR (°F)
SYSTEM TEMP IN (°F)
— AS INDICATED —
400
300
200
100
0
10.0
11.0
12.0
13.0
14.0
15.0
16.0
TIME (HRS)

Solar Kinetics & Dels - 6/21/79

TABLE 9.1

Clock Time 13:00
Relationship to Solar Noon 0.00

IN	T[1]=	85.5	P[1]=	7.80	H[1]=	26.7391
OUT	T[2]=	292.0	P[2]=	171.10	H[2]=	120.6001
TANK	T[3]=	292.0	P[3]=	171.10	H[3]=	120.6001
AMBIENT	T[4]=	94.9				
WATER	T[6]=	59.0	T[7]=	79.0		

Total Solar Flux 320 Btu/hr/sq ft
Total Solar Heat 92160.00 Btu/hr
Heat delivered by System 38805.67 Btu/hr
Estimated Heat Loss 2365.20 Btu/hr

TOTAL HEAT OF SYSTEM 41170.87 Btu/hr

Condenser Heat 35007.60 Btu/hr

Cond. Heat Balance 9.8 %

Vapor mass flow 413.44 lb/hr
Condenser Flow Rate 3.61 gpm

COLLECTOR EFFICENCY 44.7 %

$$\frac{T_{collector} - T_{amb}}{I} = .62 \ \frac{{}^{o}F\ hr\ ft^2}{Btu}$$

All temperatures in ^{o}F

All pressures in psia

program. This, along with discussions with other users of this equipment, pointed out that the phototransistors are at different rates and force the tracker out of focus at other than bright sunlight; that is, they are good only at a single light level.

9.1 Del Performance

The Del collectors were tested only briefly. This was due to the poor performance attributed to poor quality mirrors. The shipment of collectors was delayed until early spring by manufacturing difficulties. At that time, it was decided to ship them with the inferior mirrors rather than delay longer while waiting for better mirrors. Incidentally, a better quality mirror is available and Del Manufacturing has agreed to replace the mirrors if the collectors are shipped back to them. The mirrors visually appear to be bad in that an image of the receiver tube can not consistently be seen nor can any straight lines be seen in the reflections. The Del collectors were tested and the performance was much less than projected. The Del collector tested has a larger receiver tube (3/4 inch vs. 1/2 inch) than used in obtaining the efficiencies supplied by Del Manufacturing. In addition, the efficiencies supplied by Del Manufacturing are for a condition of no wind where present tests had winds varying from 5 to 20 miles per hour. These changes would have some effect on the efficiency but it is not anticipated that this would drop the projected efficiency by more than 10 percentage points. Thus, in the range tested the efficiency is projected to be no lower than about 57 percent while the tests measured 29 percent. This is about a two-to-one discrepancy. If this is to be attributed to the mirrors, then one half of the reflections must miss the receiver. Considering the visual appearance of the mirrors, this is not unreasonable. However, at least part of the error could also be attributed to the system and measurements.

9.2 Error Analysis

In order to help decipher what effect the system has on the efficiency, an error analysis based on the method of Cline and McClintock (3) was performed. This error analysis is presented in Appendix D. The results of this analysis shown the possible deviation in the measured efficiency to be 6 percent. The two largest contributors to this uncertainty were the mass flow measurement and the solar insolation measurement. The error analysis assumes there are no gross errors where data is misread, instruments which give incorrect readings, or computations are done incorrectly. In all the data taken and reduced, the data appears reasonable and consistent thereby reducing the chances for data being misread. All instruments whose output are used for data reduction have been scrutinized for error and judged to be correct. The computations are quite simple and should be correct. The only possible source of significant computational error would be in an underestimate of heat leak due to poor modeling because of uninsulated areas, etc. However, doubling the computed heat leak would increase the computed efficiency by only 2 1/2 percentage points.

In the error analysis the solar insolation accounts for half of the 6 percentage points potential error if the insolation measurement was in error by 10%. Figure 9.3 shows a plot of data taken at Barber-Nichols in Arvada, Colo. compared with data taken at NOAA in Boulder, Colo., a distance of about 15 miles. The total normal measured by Barber-Nichols and used as the solar insolation for the collector efficiency computation is in fact somewhat higher than the direct normal measured by NOAA. However, the discrepancy is 30 to 40 watt hours/meter squared out of nearly 1000 or 3 to 4 percent. This increases the measured collector efficiency by about 1 1/2 to 2 percentage points.

The results of the error analysis indicate that the discrepancy between the measured efficiency and that predicted by Del Manufacturing cannot be accounted for by measurement system error. The mirrors are left as the major source of low performance.

9.3 Solar Kinetics Performance

The Solar Kinetics data is shown in Figure 9.1. There is some appreciable scatter in the efficiency but no more than predicted by the error analysis. The efficiency with a freshly cleaned collector is about 45 percent. During the test period, the ambient temperatures were consistently in the 90 degree (F) range. The flow meter orifice was sized for the collectors to operate in the 300°F to 400°F range. This, combined with the fact that the May-June sky in Denver is clear ($I=300$ Btu/hr ft^2) or cloudy ($I < 200$ Btu/hr ft^2) with no in-between level forces the parameter on the abscissa of Figure 9.1 to fall in the narrow range of 0.6 to 0.9.

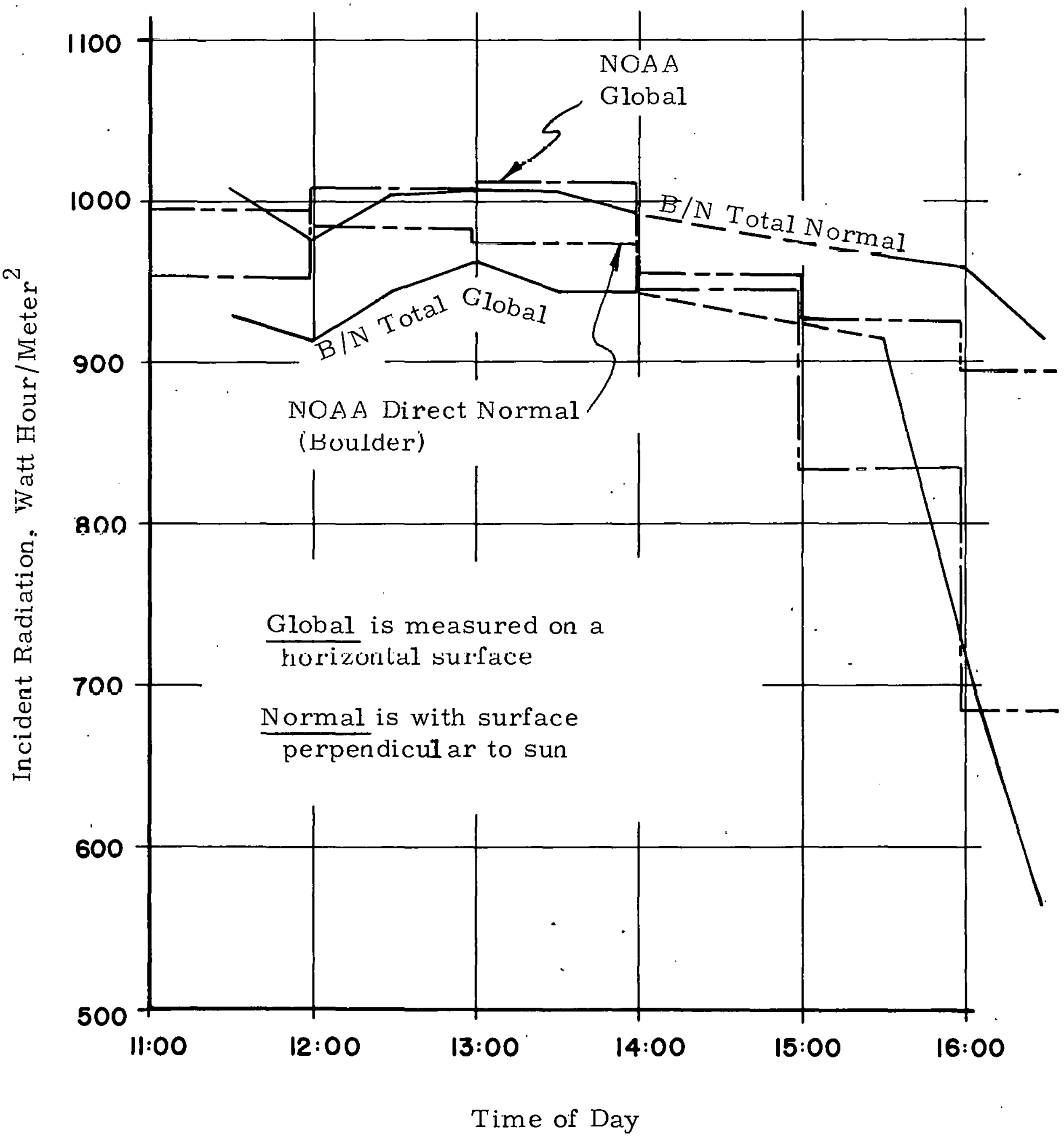
Again, the collector tested was somewhat different than the collector used by Solar Kinetics to obtain their published data. The annulus had dry air rather than a soft vacuum. Also, the receiver was a 1 inch pipe which has an outside diameter of 1.3 inches rather than a 1 inch outside diameter tube. This would reduce the predicted efficiency from that shown in Figure 9.1. In testing their collectors, the manufacturers use an oil which would exhibit a nearly linear increase in temperature as it passes through the receiver while the boiling refrigerant used here has a linear rise preheat section and a constant temperature boiling section (Figure 4.1). The manufacturer used the average temperature to define the collector efficiency and the average temperature using the refrigerant would be slightly higher thereby giving a slightly lower efficiency. It should be noted that the R-113 used here has a critical temperature of 417°F and therefore doesn't have a large constant temperature boiling section at the 350°F operating point.

The same error analysis previously discussed is valid for the Solar Kinetics data as the same system was used. Again, the solar flux used in the efficiency computation was somewhat high which in turn reduces the efficiency. When all these factors are thrown in there still would be a discrepancy of about ten percentage points.

FIGURE 9.3

Comparison of B/N Solar Radiation Measurements with NOAA.

Conclusion: The measurement of total normal made by B/N is only a few percent above the direct normal made by NOAA.



A potential source of this error that was considered was the possibility that the receiver tube wall temperature was very high due to poor heat transfer particularly in the preheat region. This would increase the receiver heat losses which would show up as a shift to the right on Figure 9.1. The average tube wall temperature would have to be in excess of 600^{o}F in order to decrease the efficiency to that measure. This type of surface temperature would cause film boiling which would reduce the heat transfer (and the mass flow) by a greater magnitude than the discrepancy measured. Thus, this high surface temperature does not seem to be the culprit.

The Solar Kinetics collectors did tend to superheat (as will be discussed below) which would elevate the average temperature and the tube wall temperature. However, the collectors were not operated in a high super-heat mode when data was being taken.

The collector receiver has a turbulator to obtain turbulent heat transfer fluid flow. This turbulator and/or the flow conditions utilized may not have been sufficient to cause turbulent flow. With oil as the heat transfer fluid, this can result in as much as 10% in efficiency points.

Thus, the discrepancy remains unsolved. The difference in the Del and Solar Kinetics efficiencies can be attributed to the quality of the Del mirrors. A means of resolving the discrepancy with the published efficiencies would be to install a valve in the liquid return line (Figure 7.6) and force the flow of liquid through the collectors to get a non-boiling efficiency with the given system. This would establish a baseline efficiency. Since neither time nor budget will allow this under the present contract, this will be proposed as one of the first steps in phase I of the follow on.

9.4 Superheating Discussion

The desired flow regime for the boil-in-collector system is to have a liquid-vapor mixture in the boiling area with liquid carryover into the receiver/separator tank. This insures a wetted tube wall and saturated conditions at the receiver tube outlet. This then provides temperature control and prevents the temperature excursion associated with dryout. There are two basic reasons why the temperature excursion is undesirable. The first is the reduced collector efficiency with increased receiver temperature and the second is the possibility of thermally decomposing the R-113 at temperatures greater than 400^{o}F.

The tests showed the Del collectors to behave in the desired manner with copious quantities of liquid carried out of the receiver tube and into the separator tank. This was identified both through sight ports at the collector exit and through the high temperature of the liquid return line (Figure 7.6). The temperature at the outlet of the receiver was nominally saturated at the pressure of the system.

The tests on the Solar Kinetics collectors were a completely different situation. At a steady state condition the Solar Kinetics collectors operated with superheat at the exit. The sight port showed a single vapor phase and the temperature was measured above the saturation temperature. During continued steady operation the temperature in the liquid recirculation line continued to cool, indicating a drop in (or lack of) recirculation rate. Figure 9.2 shows the temperature into the collector dropping while the collector temperature and the system-in temperature remain nearly constant. The degree of superheat seems to be a function of the temperature level of the system. Although the system was not tested over a wide range of temperatures (275^{o} to 350^{o}F), the superheat did tend to increase, being near saturation at 275^{o} to 300^{o}F, 20^{o}F superheat at 325^{o}F boiling and more than 50^{o}F superheat at 350^{o}F. The collectors were protected with a temperature limiting switch that moved them out of focus at temperature greater than 400^{o}F. Thus, data above 350^{o}F boiling was not obtained.

The superheat variation was also a function of the feed pump operation. While the feed pump was on, the superheat decreased and sometimes went to saturation (at lower boiling temperatures). When the pump cut off, there was some overshoot (continued drop in temperature) due to delay in the flow system and then the superheat would begin to increase. This would continue until the pump cut back in. The temperature swing was often in the 20^{o}F range. Looking at figure 7.6, the feed pump supplies liquid to the lower header. The path the fluid takes only depends on the relative flow resistance through the collectors compared to the flow resistance through the liquid recirculation line. These resistances should be reasonably balanced with probably less resistance (due to larger lines) in the liquid recirculation line. Therefore, some flow would be forced up the receiver tube, bringing it back toward saturation.

The superheating in the Solar Kinetics collector but not in the Del collector can be tied to the requirement of the Solar Kinetics collector to rotate the receiver tube through an arc. This makes it necessary to use a flexible tube to allow this motion. Figure 9.4 shows the elevation view of the two different collector systems. The non-rotating receiver tube of the Del collector makes it possible to have a continuous uphill piping system through the upper header and up to the tank. This essentially duplicates the successful laboratory test system. The Solar Kinetics system was installed with the flexible line such that at the solar noon position the receiver outlet is about 18 inches above the header. This apparently creates a manometer resulting in a vapor block in the receiver tube. The riser from the header to the tank fills with liquid and blocks the receiver flow. Changing the plumbing on the upper end of the Solar Kinetics to eliminate this dog-leg would correct this problem.

An interesting feature of this system is shown in Figure 9.5. The liquid recirculation line typically has lower temperature liquid (high density) while the preheat and boiling receiver tube has lower density liquid and vapor. There-

FIGURE 9.4

COLLECTOR PLUMBING
ELEVATION VIEW

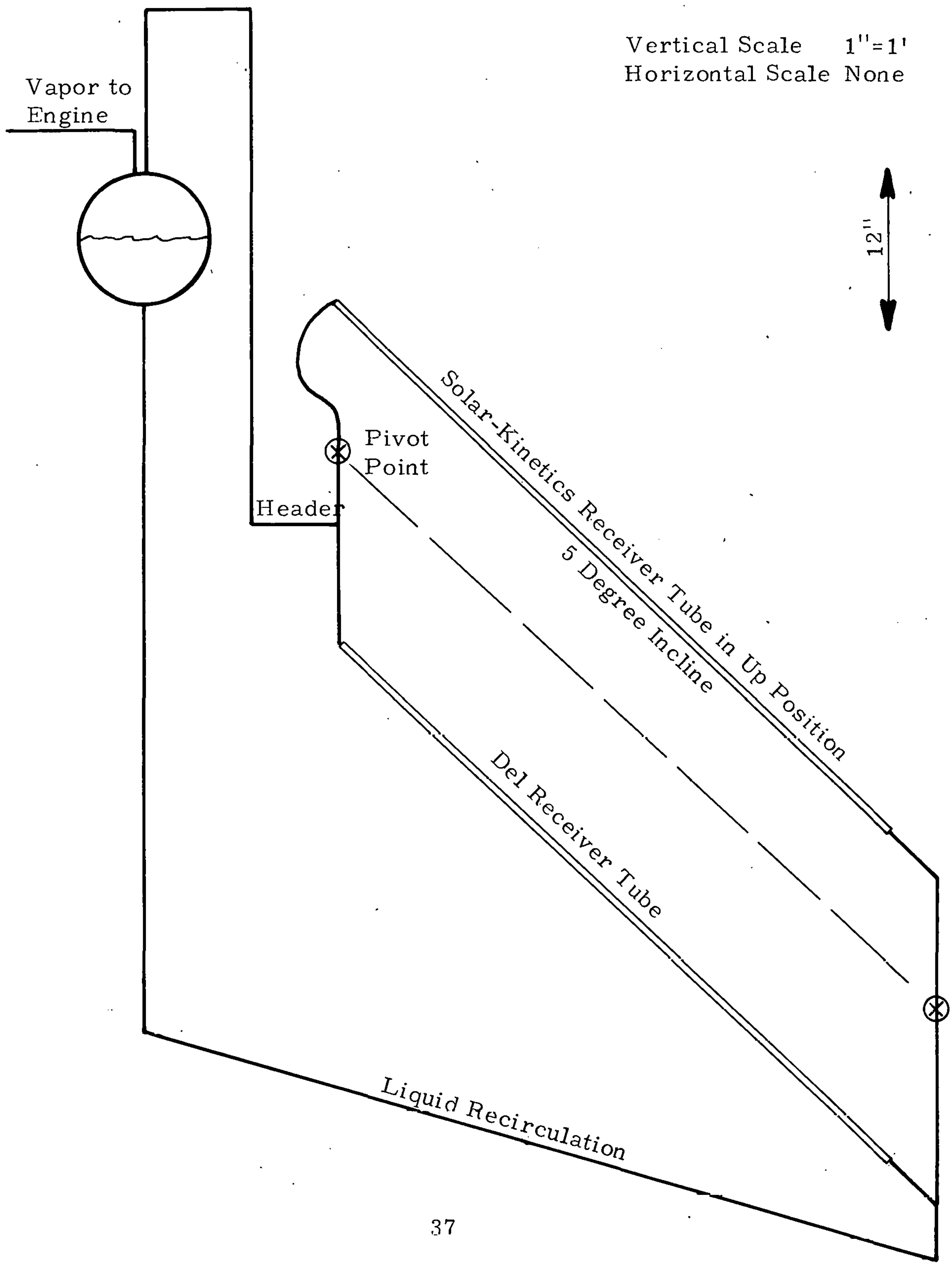
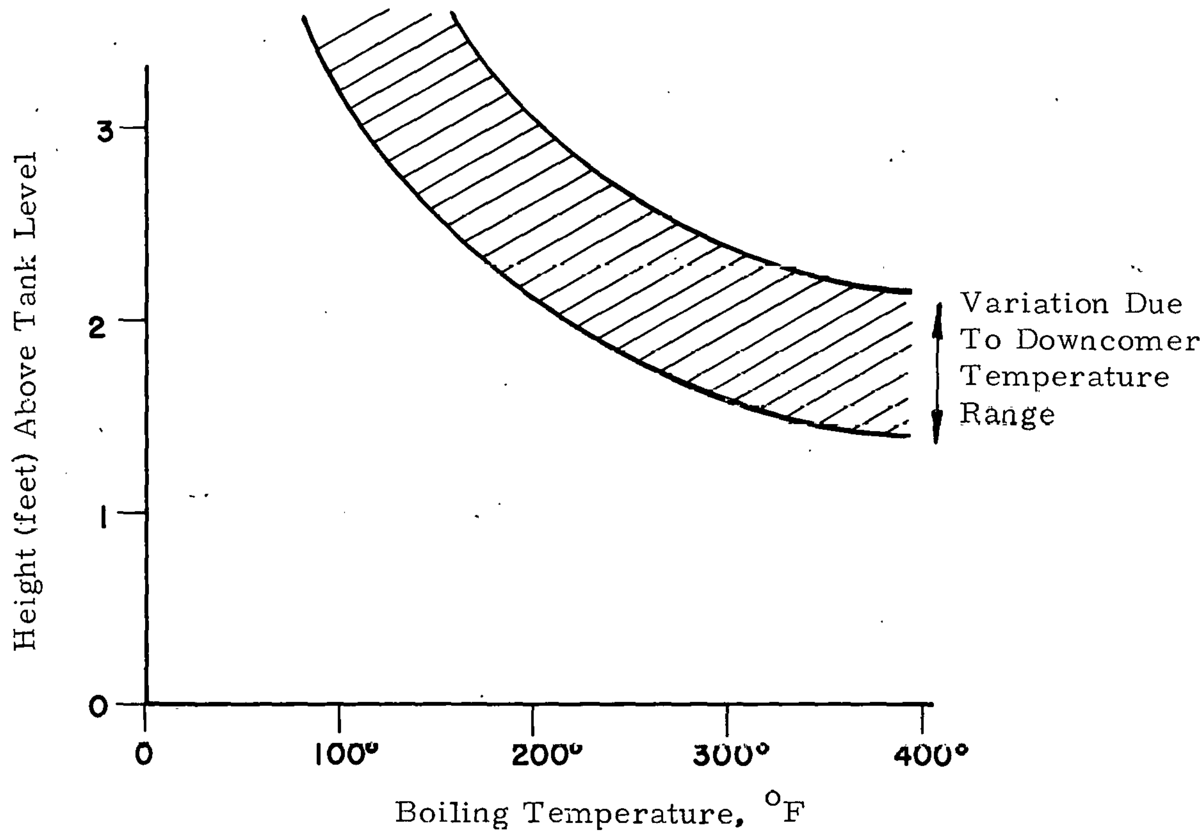
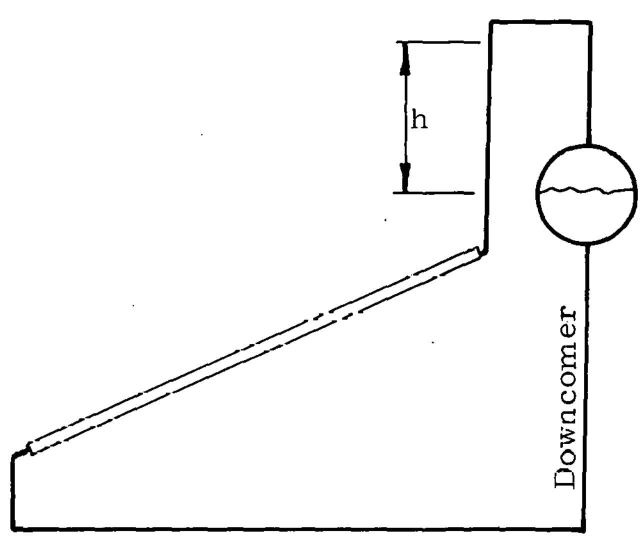


FIGURE 9.5

CARRYOVER HEIGHTS





The above graph depicts the height above the tank level (h) that the low density (hot-boiling) R-113 will exist due to the counter-balancing column of high density (cooler) R-113 in the downcomer.

fore, a simple manometric analysis shows the lighter leg will balance in the order of 2 feet above the liquid level in the tank. This would always be sufficient to insure a wetted tube if there was not a flow blockage.

9.5 Combined System Performance

The combined systems of Del and Solar Kinetics were tested on two occasions. The efficiency was both lower and higher than expected. The Del collectors low efficiency and smaller surface area gives them a total heat contribution of 35% compared to Solar Kinetics 65% or almost a 2 to 1 ratio. Therefore, it would be expected that the combined systems would operate at an efficiency of near 40%. Instead, the numbers were 31% and 45%. The latter figure was considered more reliable as the system was operated longer for more steady operation and the system had been freshly cleaned. During the combined testing, the high liquid carryover rate from the Del collectors choked off the riser from the header to the tank to a greater extent than with the Solar Kinetics alone. This forced the Solar Kinetics collector to superheat more than it would have without the Del collectors operating. This is consistent with the previous analysis.

9.6 Heat Balance

The heat balance on the condenser showed the heat delivered in the refrigerant to be about 10% more than the heat picked up in the water. While this is a difficult measurement to make, it is consistent as any heat lost to the ambient from the uninsulated jacket would force the heat balance in this direction. The heat balance did vary but there was no discernable pattern between heat balance and efficiency. Generally, the heat balance proved the heat measured as delivered by the system.

9.7 Summary

Due to the undesirability of testing the poor performing Del collectors and the superheating problem with Solar Kinetics, it was decided not to change the inclination angle from the preset 5 degrees. It would appear that Del would operate at less than 5 degrees but since the heat flux is low due to the poor mirrors, the results would be in question. To find the minimum angle it is necessary to go past that angle to where the flow stops and superheating begins. Because of the operation of the Solar Kinetics it would be impossible to tell this effect from the flow blockage superheat problem.

In summary, the collectors do boil as anticipated but some unexpected problems were encountered. The superheat problem can be corrected by a change in the plumbing and the low performance of the Del collectors can be corrected with new mirrors. The low performance of the Solar Kinetics needs more testing (with a non-boiling fluid) in order to diagnose the problem or establish a better baseline to measure from. These items are suggested in the proposed follow-on.

10.0 CONCLUSIONS

The lack of time and funds made it necessary to terminate the work before some critical issues were resolved. Therefore, conclusive decisions cannot be made at this juncture. However, the evidence does indicate that:

1) The concept is proven. The system operates in a very stable manner with no control problems, even with the very simple system used.

2) The continually upward piping system as used with the Del collectors is a necessity. Such a system needs to be devised for collectors that do not rotate about their receiver tube.

3) Boiling at a 5 degree inclination does work. The efficiency problem is unresolved.

4) The collector system does provide a saturated vapor suitable for turbine feed.

5) The tracking system employed did not prove capable of unattended operation, and would need extensive rework to do so.

6) The fluid level controls, general operating controls, and other ancillary equipment proved very satisfactory.

7) The discrepancy in measured efficiency with that claimed by the manufacturer is unresolved. If this were due to the boiling-in-collector technique because of high surface temperatures, this would tend to negate the performance benefits.

11.0 RECOMMENDATION AND PROPOSED FOLLOW ON

There are some obvious questions that must be resolved. Once they are resolved the next step leading toward a reliable (commercial) system that could be applied to many areas of use would be the demonstration of the system by a full year's operation. This would demonstrate reliability, establish maintenance requirements and procedures, and determine the total energy delivered during an operating year. In order to make such a demonstration meaningful, it would be on a system of such size as to require a full bank of collectors manifolded together to test parallel operation. Further, more collectors would better test the tracking and driving mechanisms. The larger field (1000 sq ft) would also provide sufficient energy to drive an engine large enough to provide significant useful work. Therefore, the proposed follow up work breaks into two distinct phases with a clear cut decision point separating them. A successful conclusion of both phases would provide operational information that has not yet been obtained from any of the systems previously built.

11.1 Scope of Work

Phase I

a) Replace the Del Manufacturing mirror modules with mirror modules of higher quality.

b) Rework the exit plumbing on the Solar Kinetics to eliminate the vapor trap.

c) Test the collectors in an all liquid (non-boiling) mode to establish a base line efficiency for each of the collectors in conjunction with the system.

d) Retest the Del collectors in a boiling mode to determine the improved efficiency due to the new mirrors and compare with the baseline efficiency.

e) Retest the Solar Kinetics collectors to prove out the new plumbing system and verify the efficiency in comparison with the baseline efficiency.

f) Prepare a final report on this work as information for the decision on moving into Phase II.

Phase II

g) Complete the turbo gearbox and install in the system.

h) Obtain and install an induction electric motor with necessary instrumentation to load (as a generator) the turbine and determine the total energy delivered. Install suitable engine and collector controls and other ancillary equipment for continuous, automatic operation.

i) Select, purchase, and install the collectors necessary to expand the field to approximately 1000 square feet.

j) Operate the system for one calendar year during which the system will be monitored on a daily basis to ensure proper operation. Daily performance, maintenance requirements, and problems encountered will be recorded.

k) Prepare a detailed boil-in-collector engine system cost analysis for comparison with the conventional oil system. This analysis will include installation costs.

l) Prepare reports on the above work.

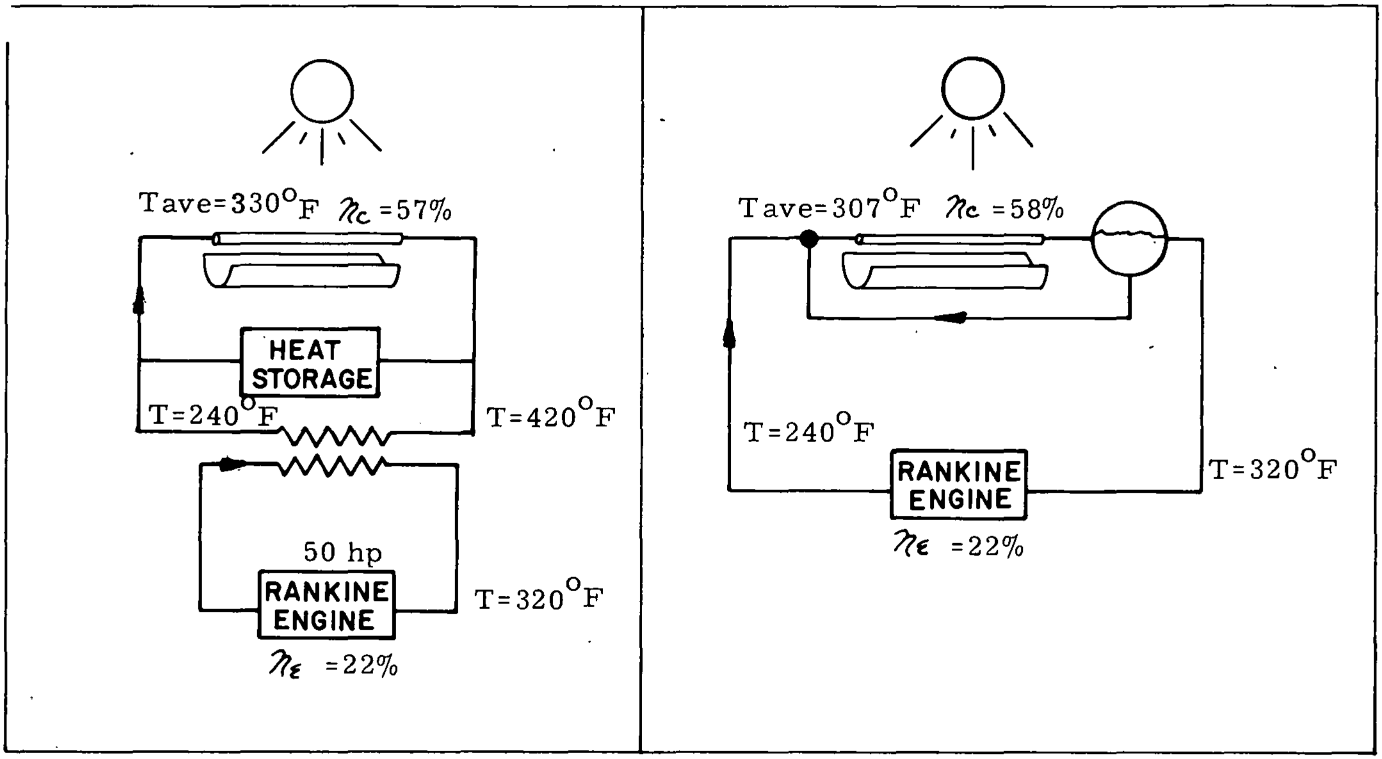
12.0 BIBLIOGRAPHY

1) R.E. Barber, "Solar Powered Organic Rankine Cycle Engines - Characteristics and Costs", Eleventh Intersociety Energy Conversion Engineering Conference, AICHE, 1976, pp. 1151-1156.

2) R. Alvis, Private Communication , April 19, 1978.

3) S.J. Kline and F.A. McClintock, "Describing Uncertainties in Single Sample Experiements", Mechanical Engineering, p. 3, January 1953.

APPENDICES



OIL HEAT STORAGE

BOILING IN COLLECTOR, B.I.C.

ASSUME:

A) Solar Insolation = 1551 BTU/ft^2 Day, Day = 10 hrs:

B) For Ac=7000 ft^2, Oil has 3 HP Parasitic and B.I.C. has 1HP Power Varies Linearly With Ac

C) Rankine Cost \$500/HP For Oil, \$400/HP For B.I.C.

D) Collectors Cost \$10/ft^2

E) Foundation Cost \$.72/ft^2

F) Cost Info From April 19, 1978 Memo By R. Alvis

1) generated $\dfrac{\text{HP-HR}}{\text{Day}} = \text{Acft}^2 \ \dfrac{1551 \quad \text{BTU}}{2545 \ \text{BTU ft}^2} \ {}^2 \ \text{Day} \quad \eta c \ \eta e \quad \text{hr HP}$

$$= .134 \ \text{Ac}\,\eta\text{c} \ \frac{\text{HP-HR}}{\text{Day}}$$

2) operating hrs $= \dfrac{\text{HP-HR}}{\text{Day}} \quad 50 \text{ HP}$ Oil

$\text{HP} = \dfrac{\text{HP-HR}}{10 \text{ HRS}}$ B.I.C.

3) parasitic $\dfrac{\text{HP-HR}}{\text{Day}} = (3)(\text{operating hrs}) \ \dfrac{\text{AC}}{7000}$ Oil

parasitic $\dfrac{\text{HP-HR}}{\text{Day}} = (1)(10)(\dfrac{\text{AC}}{7000})$ B.I.C.

4) assume $C_p = .65$ Btu/lb$^\circ$F $\rho = 6$ lb/gal
$\Delta T = 420-240 = 180^\circ F$
then to keep a 50 hp, 22% engine running

$$\text{heat storage gal} = \frac{(50\text{ hp})(2545)\text{ BTU}}{(.22)\quad\text{hr}} \quad \frac{\text{lb}^\circ\text{F(opt. time-10hs)gal}}{.65\text{ BTU}\ \ 180^\circ\text{F}\ \ 6\text{lbs}}$$

$$= 824\,\frac{\text{gal}}{\text{hr}}(\text{opt. time-10})\text{ hrs}$$

5) Piping costs = \$.17/ft^2 both systems

6) valves = 3000 + \$.083/ft^2 oil
= 1000 B.I.C.

7) expansion tank = 7000 gal \$.35/gal oil
20,000 gal \$.18/gal
= \$100 B.I.C.

8) Thermal storage @ \$3 gal

9) R-113 fluid = .016 gal/ft^2
@ 10 lb/gal and \$.50/lb
R-113 fluid = \$.08/ft^2

Appendix A-3

COMPARATIVE COST ANALYSIS
TABULAR COMPUTATION

	12,000 ft^2		15,710 ft^2		9,330 ft^2		6546 ft^2	
2)	OIL 18.3 hrs @ 50 hp	BIC 10 hrs @ 93.3 hp	OIL 24 hrs @ 50 hp	BIC 10 hrs @ 122 hp	OIL 14.2 hrs @ 50 hp	BIC 10 hrs @ 72.5 hp	OIL 10 hrs @ 50 hp	BIC 10 hrs @ 51 hp
1) Generated HP-HRS/Day	916	933	1,200	1,221	713	725	500	509
3) Parasitic HP-HRS/Day	94	17	162	22	57	13	28	9
NET HP-HRS/Day	822	916	1,038	1,199	656	712	472	500
4) Heat Storage Gal	6,839	-0-	11,540	-0-	3,460	-0-	-0-	-0-
Heat Engine	25,000	37,320	25,000	48,800	25,000	29,000	25,000	20,400
Collectors	120,000	120,000	157,100	157,100	93,300	93,300	65,460	65,460
Foundation	8,640	8,640	11,300	11,300	6,700	6,700	4,700	4,700
5) Piping	2,040	2,040	2,700	2,700	1,600	1,600	1,100	1,100
6) Valves	3,996	1,000	4,300	1,300	3,800	800	3,500	600
Foundation - Equip.	2,000	2,000	2,000	2,000	2,000	2,000	2,000	2,000
Electrical Wiring	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000
7) Expansion Tank	2,400	100	3,500	100	1,400	100	100	100
Auxiliary Pumps	3,000	1,000	3,000	1,000	3,000	1,000	2,000	1,000
8) Heat Storage Oil	20,500	-0-	34,600	-0-	10,380	-0-	500	-0-
9) R-113	-0-	960	-0-	1,260	-0-	746	-0-	520
TOTAL	192,580	178,060	248,400	230,560	152,180	140,250	109,364	100,880
\$/HP-HR/YR	.642	.533 17%	.656	.528 20%	.540 15%		.635	.553 13%

Appendix B-1

SOLAR POWERED RANKINE CYCLE IRRIGATION PUMP: MARKET SURVEY OF EXISTING IRRIGATION PUMPS AND ANALYSIS OF PUMP MATCHING WITH RANKINE ENGINE

Contract No. ET-78-C-03-1891

November, 1978

Prepared For: Department of Energy

Prepared By: Barber-Nichols Engr. Co.

6325 West 55th Avenue, Arvada, Colorado 80002

Appendix B-2

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Market Survey and Selection of the Engine Size for Phase II Laboratory Test

The original proposal by Barber-Nichols was made in conjunction with Soltrax, Inc. in which Soltrax would provide the Rankine engine hardware and perform the market survey. Subsequently, Soltrax withdrew from their solar irrigation activities and sold the prototype Rankine cycle (R/C) engine hardware to Barber-Nichols. Therefore, the market survey was changed in nature from a survey designed to select a specific market area and Rankine engine size to suit a specific sales company (Soltrax) to a more general study consistant with the overall goals of DOE and Barber-Nichols.

1.0 CURRENT STATUS OF IRRIGATION PUMPING IN THE U.S.

Currently, there are hundreds of thousands of irrigation wells in use throughout the United States. Irrigation pumps range in size from a few horsepower to several hundred horsepower and are powered by natural gas, electricity, liquefied natural gas, and diesel fuel. In the states of Arizona, New Mexico, Texas, and California alone - representing 43% of all the irrigated land in the U.S. - there are nearly 70,000 natural-gas-powered irrigation pumps. Approximately 100,000 electrically-powered irrigation pumps are in use in California, (Ref. 1).

U. S. farmers irrigated over 35 million acres in 1974 with 69 million acre feet of water pumped from wells and surface water. Acres irrigated by type of energy used to pump the water were estimated at 15.6 million for electricity, 10.6 million for natural gas, 3.9 million for diesel, 3.3 million for LPG, and 1.5 million for gasoline. Energy consumed was estimated 19 billion KWH, 132 billion cubic feet of natural gas, 178 million gallons of diesel fuel, 237 million gallons of LPG, and 71 million gallons of gasoline. The combined direct energy in these fuels equals 260 trillion Btus (does not include the Btus required to generate the electricity). This represents about 20% of all energy used on farms for production of commodities and livestock, (Ref. 2).

An estimated $594 million was spent in 1974 for energy for on farm pumping of irrigation water. The least expensive source of energy for pumping was natural gas followed by electricity, diesel, LPG, and gasoline. California uses the most electricity among all states for irrigation energy while Texas used the most natural gas, Nebraska is the largest user of diesel and LPG, and Arkansas is the largest user of gasoline, (Ref. 2).

Energy expenditures in 1977 for farm irrigation pumping are estimated to exceed $800 million for all of the U.S., with approximately $700 million of the total in the 17 Western states. Energy costs from conventional fuels are projected to increase dramatically over the next decade due to greater demand, increasing lift requirements, shortages of natural gas, and rising prices of fossil fuels and electricity. In some regions of the U.S., farming has already become uneconomic due to rapidly increasing costs of pumping irrigation water. In view of these increased energy costs, solar energy is now receiving attention as a potential alternate source of energy for irrigation and other agricultural applications (Ref. 1).

Over 80% of the pumped irrigated lands and 90% of the pumping energy is shown in Table 1 and Figure 1 (from Ref. 2) to be in the Western half of the nation where sunshine is abundant, making this the prime area for application of solar pumping systems. Four states - Texas, Nebraska, California, and Kansas - make up approximately 59% of the irrigated acreage and use 57% of the pumping power (Table 2). These values increase to 81% and 70%, respectively, if only five more states are added to the list to make a total of nine states. This concentration of use simplifies selection of the market area for solar pumps.

Unfortunately, detailed information on the pumping requirements for all nine of these states is not available. Reference 2 is the most complete compilation, but the date base is 1974 and much calculation was necessary to complete the lists. More current information for several states of interest is shown below.

1.1 <u>Nebraska - 12.1% of energy usage - information from Reference 3</u>

Nebraska has over 52,000 irrigation wells supplying water to over 5.5 million acres of land. Pumping water for irrigation is highly energy intensive, requiring 53 gallons of diesel fuel per acre for a typical center pivot irrigation system and 31 gallons of diesel fuel per acre for a typical gated pipe irrigation system. For a center pivot, the energy cost per acre is ten times the energy cost of the cultural and harvesting operations. On a statewide basis, irrigation accounts for nearly half of all energy invested in production agriculture.

Because of excellent water resources and the technical development of the center pivot irrigation system, irrigated acreage in Nebraska has increased from 4.1 million acres in 1968 to 6.3 million acres in 1976, accounting for 25% of total lands added to irrigation in the U.S. during this period of time. There were over 11,750 center pivots operating in Nebraska in 1976, as identified by satellite imagery.

At present, 23% of the irrigation pumps are powered by natural gas; 25% are electrically powered; 14% are powered by LPG; and 37% are powered by diesel engines. With limitations on expansion of electrical power and natural gas and the high cost of LP gas, almost all new systems are diesel powered. Since 1968 the proportion of diesel powered systems has increased from 23% to the present 37% and this trend can be expected to continue.

The exhaustion of natural gas and petroleum resources will place great stress on food and fiber production, especially in those areas relying on irrigation from pumping plants. An economic alternative energy source must be developed before economics force major agricultural areas out of production.

1.2 <u>California - 6.4% energy usage - information from Reference 4</u>

For water studies, California has been divided into nine different hydrologic basins. This report presents data for the six most important basins.

California irrigated acreage is about 9,099,000, consisting of 6,186,000 acres of field crops, 1,837,000 acres of trees and vines, and 1,076,000 acres of vegetable crops. The total amount of water applied annually for irrigation in California is about 32,000,000 acre/feet, an average of about 3.5 acre/feet per acre. Consumptive use estimates for the state are approximately 20,000,000 acre/feet per year. The difference occurs because some return flow water is repeatedly reused if quality permits. Some is nonrecoverable because of position or poor quality.

Energy costs for different alternative supplies of water in the different portions of the state range from 34 KWH per acre/foot for some of the areas where gravity water is supplied in canals to 3,000 KWH per acre/foot for water supplied by the State Water Plan that is developed in the northern part of the state, transmitted 500 miles to the south through canals and pipes, and lifted 3,000 feet over a mountain range.

1.3 New Mexico - 8% of energy usage - information from Reference 5

There are in excess of 10,000 irrigation pumps in New Mexico. More than half of them are presently powered by natural gas. The price is increasing rapidly and availability is declining for this use of gas. New energy sources must be found if New Mexico farmers are to survive.

An evaluation of the present numbers and sizes of farm stationary power plants was made by Mr. Robert Alvis of Sandia Laboratories (Ref. 6) and is shown in Table 3 for the states of Texas, California, Washington, Idaho, Arizona, New Mexico, Nevada, and Utah. This table shows that a total of 295,000 engines at an average size of 80 hp are now in use. Assuming a ten year engine life, a replacement market for 30,000 engines per year presently exists in these eight states. If solar units could capture 10% of this market, 3,000 units per year, at a specific cost of \$15,000 per installed KW (Ref. 6) a market of \$2.5 billion per year exists. These figures should interest any large manufacturing company.

On an energy basis, Sandia Labs points out in Reference 7 that in 1974 the energy requirement for pumping water was enough to heat 5 million homes. This number has undoubtedly increased substantially in the past four years. Mr. Newkirk points out in Reference 8 that,"it is estimated that replacement of the over 160,000 natural gas powered irrigation wells with solar powered pumps in California, Arizona, New Mexico, and Texas would create an energy saving of 1.4×10^{14} Btu's or 1.4×10^{11} cfm of natural gas annually based on 1969 data".

In conclusion, 74% of the irrigation pumping energy requirements occur in areas with abundant solar insolation, namely Texas, Nebraska, Kansas, New Mexico, California, and Arizona. The current source of this energy is 25% electricity, 10% diesel, 3% gasoline, 53% natural gas, and 9% LPG. The future availability of these fuels for irrigation purposes is in question. Therefore, the potential market for a cost effective solar pumping system is very large. It is estimated that 295,000 irrigation engines at an average power level of 80 horsepower were in use in 1977. If only 1% of these engines were replaced with solar pumping systems each year a market of $2.5 billion per year results. The key problem is obtaining a cost effective solar pumping system.

2.0 HEAD-FLOW CHARACTERISTICS

An evaluation of head-flow characteristics of a typical well is being made to determine the power level of a typical pump drive and to determine if available pumps can match this requirement. Matching the engine to the requirement presents no problem as shown in Figure 2. This figure shows that typical deep well pumps, namely Crane-Deming units, cover essentially all heads and flows at efficiencies in excess of 78% in the hp range of 10, 25, and 50. Any desired power level can be covered with this line of pumps. The nomenclature shown on this curve indicates the pump model (the number of stages of that model to obtain the specific flow and head).

2.1 Head Characteristics

Considerable information is available on irrigation pump head requirements. Reference 2, for example, lists the average lift of both surface and ground water for each state in the U.S. This information is presented in Tables 1 and 2 for regions and states utilizing the most irrigation pumping energy. As shown in Table 2, ground water requires from 100 to 350 feet of lift while surface water requires only 5 to 40 feet.

In addition to the well pumping lift required, the pump must provide pressure for the specific distribution system used on the farm. Table 4 (from Ref. 9) shows the distribution system pressure losses for various approaches. As shown here the pressure loss can vary from approximately 1 psi for an open ditch without returns to as high as 110 psi for various mechanical moving types and big gun types of distribution systems. The pressure required for center pivot units with water power drive runs in the neighborhood of 80 to 90 psi. However, low pressure systems reduce this loss to as low as 30 psi. It is obvious that with a solar irrigation system a low pressure loss distribution system is a necessity in order to utilize the bulk of the solar energy for pumping (head that can't be controlled) rather than distribution (head that can be controlled). Reference 9 provides considerable information on ways to reduce the distribution pressure loss and balance irrigation water requirements so that the pumping system can be utilized over a longer period of the year by crop mix and other techniques.

2.2 Flow Characteristics

Although there is adequate information on head requirements, unfortunately there is little flow information for the typical irrigation pump. Reference 10 is an exception since it does show the flow and head ranges and averages for the nine major counties in New Mexico utilizing irrigation pumps. This information is presented in Figure 3, which shows that pumps in New Mexico covered a rather wide flow range, and no specific trend is shown. However, it is interesting to note that approximately 85% of the wells can be handled with an approximate 40 hp capacity pump. For reference the center pivot type of irrigation system is commonly sized for approximately 130 to 140 acres and utilizes between 650 and 1000 gpm on this amount of land.

2.3 Horsepower Characteristics

The head-flow characteristics of a well are defined; however, determining the size of the typical pump is considerably more difficult since it is a function of the specific well, crops, distribution system, terrain, and required rate of flow. It is interesting to note, however, that for typical New Mexico wells (Figure 2), a 25 hp pump would meet the needs of approximately 36% of the wells and a 40 hp pump would meet the needs of 85% of the wells (not including distribution system losses). In a study of the 1976-77 irrigation market made by a diesel manufacturer, it was estimated that 43% of the pumps purchased would be in the 0-50 hp range, 28% in the 51-100 hp range, 22% in the 101-150 hp range, 5% in the 151-200 hp range, and 1% in the greater than 200 hp range. Unfortunately, a break down of the 43% segment less than 50 hp was not made. This 0-50 hp pump size is the area which is of prime interest in the solar powered irrigation regime.

Since pump power is a function of head and flow which is dependent on the specific well, the crop being irrigated, the distribution system, and the terrain, it is impossible to define the nominal power level engine size for a typical solar powered irrigation system. The marketable size will be selected on other criteria probably unique to the manufacturer developing the market. The Rankine cycle engine should be as large as possible to reduce the specific cost per horsepower. However, the collector field should be small to fit the available land and reduce system complexity. Based on the information provided herein it is estimated that the practical engine size is in the 20-50 hp range.

In conclusion, a) the typical well head varies from 5-40 feet for surface water and 100-350 feet for ground wells, b) distribution pressure losses for solar pumps should be limited to 1-30 psi, c) the flow rate for a typical center pivot unit covering 140 acres is approximately 600 gpm, d) there is a sufficient quantity of high efficiency deep well pumps available for coupling to any size solar engine, e) 20-50 hp solar pumps would meet a substantial number of irrigation needs, f) the marketable system size will depend on factors other than head-flow and probably will be selected by the manufacturing company to meet requirements in certain areas of the country.

3.0 ELECTRIC VERSUS DIRECT DRIVE OF PUMP

In evaluating the marketable pump three possible drive systems may be selected. These include the electrical only, direct shaft drive only, and shaft drive with electrical standby. Each of these have advantages and disadvantages as discussed below.

3.1 Electric Drive Only

This system is composed of a solar Rankine power system which generates electricity and puts it into a private or public grid for distribution to several irrigation pumps throughout the area. The advantages of this system are: 1) the power system can be remote from the pumping locations, 2) flexibility is unlimited in the location of both engine and pump, 3) startup of the engine and pump would be easy since both can operate independently from the other while public utility power is used as standby power, 4) the solar device can be paralleled with the public utility grid, 5) the power grid acts as the energy storage device since surplus energy is fed into the grid and then withdrawn at a later time when needed.

The disadvantages of this system are: 1) it is lower in efficiency due to a Rankine generator efficiency of 85%-95% (depending on size) and motor efficiency at the well head of 50%-95% (depending on the configuration), 2) pump well water may not be available for Rankine engine cooling, and 3) power lines must be provided both to the well site and to the solar generating site.

In general, the electric-only system has the maximum flexibility; however, it has the lowest efficiency of the three approaches.

3.2 Direct Shaft Power Only - No Standby Motor

This system is a free standing unit supplied with very low amperage public power for use by the controls and during startup. Advantages of this system are: 1) it is the most efficient since no electrical losses occur in the drive train, 2) the pump and engine can run at variable speed so that the pump automatically loads the engine resulting in load control, and 3) there is negligibly low purchased power. Disadvantages are: 1) the Rankine engine must be located directly coupled to the pumped well, 2) a complex water reservoir is necessary to provide water to the Rankine condensers during startup, and 3) there is no standby well pumping power.

This is the most efficient system but has little flexibility, which the opposite of the electrical-only system.

3.3 Shaft Drive with Electrical Standby

This system has a direct shaft drive from the Rankine engine to the well pump. A clutch would be provided between both the standby electric motor drive and the Rankine drive to allow operation of either one or both of the power sources. This system can be designed to have all the advantages of both the electrical-only and the direct shaft drive-only systems, however, it does incorporate some disadvantages of both systems, namely: 1) power must be supplied to the Rankine engine and any remote pumps to be powered by the solar system, and 2) the Rankine engine must be hooked to one of the wells which operates whenever the solar engine system is running.

This system has all the advantages of both the electric only and the direct shaft drive only with only a few of the disadvantages.

3.4 Summary of Pump Drives

The basic differences in the Rankine engine utilized with any of the three systems above are in the control and startup areas only. Therefore, a unit can and should be designed which can be applied to any of the three approaches in order to supply the largest market. It is felt that the third system, shaft drive with electrical standby, is the superior system and should be used in most applications. However, all three approaches have good and bad features. Consequently, a prototype machine should be flexible enough to operate in all three modes.

4.0 PLAN TO MEET MARKET NEEDS

The original marketing plan presented to DOE in the Barber-Nichols proposal involved Barber-Nichols doing the engineering research and development of the solar irrigation system and Soltrax, Inc. doing preproduction marketing sales studies to result in selection of a unit size for follow-on phases. Because of Soltrax's retreat from the solar irrigation field, the present marketing plan for solar irrigation is the same one Barber-Nichols currently utilizes for its development of solar air conditioning systems. Specifically, Barber-Nichols will do engineering design, development, and field testing and will supply government needs in preproduction quantities. When a commercial market develops and production quantities are feasible, Barber-Nichols will team with other companies having capabilities in the areas of manufacturing, sales, and field service to meet the commercial market.

Solar powered systems are not economically feasible at the present time. Thus, the commercialization plan leading to specific market appeal has not been formulated. However, the Barber-Nichols management is committed and the Barber-Nichols staff is qualified to supply one production engine and quantities up to, perhaps, 100 units per year for government use.

5.0 PROPOSED ENGINE DEVELOPMENT PLAN

The proposed plan to develop a solar irrigation engine for field applications is summarized below.

5.1 Phase I

Phase I, the proof-of-the-concept phase, will be completed. The boiling-in-the-collector concept will be demonstrated in a 300 ft^2 collector field and the Rankine engine will be operated.

5.2 Phase II

The present engine will be coupled to 300 ft^2 of collectors and controls will be developed so that the unit can operate on an automatic basis whenever the sun shines. A one or two month test at 3 hp turbine shaft (approximately 1.5 hp output shaft) will be conducted. The duration of Phase II will be four months.

5.3 Phase III

A typical collector "wing" (1000 ft^2) will be tested. 700 ft^2 of the superior collector based on Phase I Barber-Nichols tests will be purchased and installed with the existing 300 ft^2 at the Barber-Nichols facility. The present Rankine engine will be modified to run at 10 hp, connected to the 1000 ft^2 collectors, and will run for 5-6 months to obtain operating experience. The duration of this phase will be eight months.

5.4 Phase IV

Field models of 10-50 hp solar irrigation systems will be designed and built. The size and number will depend on government requirements.

6.0 REFERENCES

1) "Solar-Powered Irrigation Systems", brochure by Battelle Memorial Institute, 1977.

2) Sloggett, Gordon, "Energy Used for Pumping Irrigation Water in the United States, 1974", presented at the Solar Irrigation Workshop, Albuquerque, NM, July 7-8, 1977.

3) Splinter, Dr. W. E., "Needs for Solar Irrigation - Nebraska", proceedings of the Solar Irrigation Workshop, Albuquerque, NM, July 7-8, 1977. Available from NTIS.

4) Marsh, Dr. A. W., "Water Use in California and Energy Requirements of Alternative Sources", proceedings of the Solar Irrigation Workshop, Albuquerque, NM, July 7-8, 1977. Available from NTIS.

5) Abernathy, Dr. G. H., "ERDA/New Mexico Experiment", proceedings of the Solar Irrigation Workshop, Albuquerque, NM, July 7-8, 1978, Available from NTIS.

6) Alvis, Robert, private letter of May 31, 1978.

7) Alvis, R. L. and Lukens, L. L., "Solar Irrigation Program Plan, Second Revision", Sandia Labs report SAND/78-0308, May, 1978.

8) Newkirk, H. W., "Solar Technology Applications: A Literature Review of Solar Powered Irrigation Systems", Lawrence Livermore Labs report UCID-17510, June 30, 1977.

9) "Agricultural Practices Which Could Enhance Solar Powered Irrigation Plant Utility", report to Sandia Labs by the University of Arizona - Sandia agreement No. 07-6915, report period Sept., 1977-May, 1978.

10) Long, Roger L., "Cost of Pumping Irrigation Water in Ten New Mexico Counties", New Mexico Agricultural Experiment Station Bulletin 490, March, 1960.

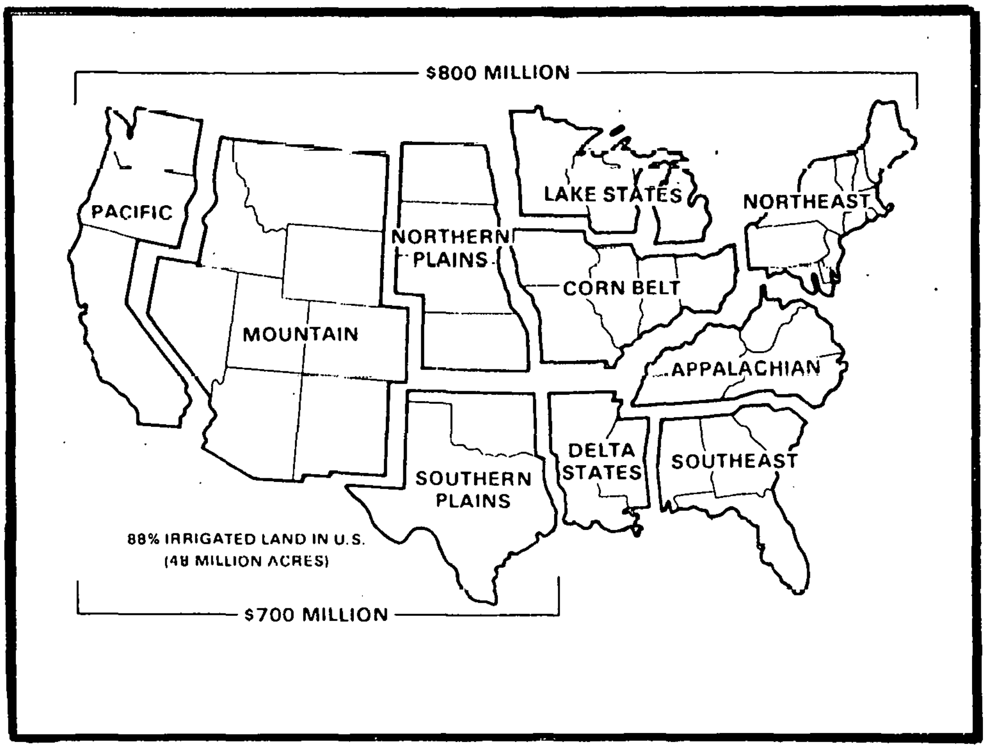


Figure 1. Estimated 1977 energy expenditures for farm irrigation pumping.

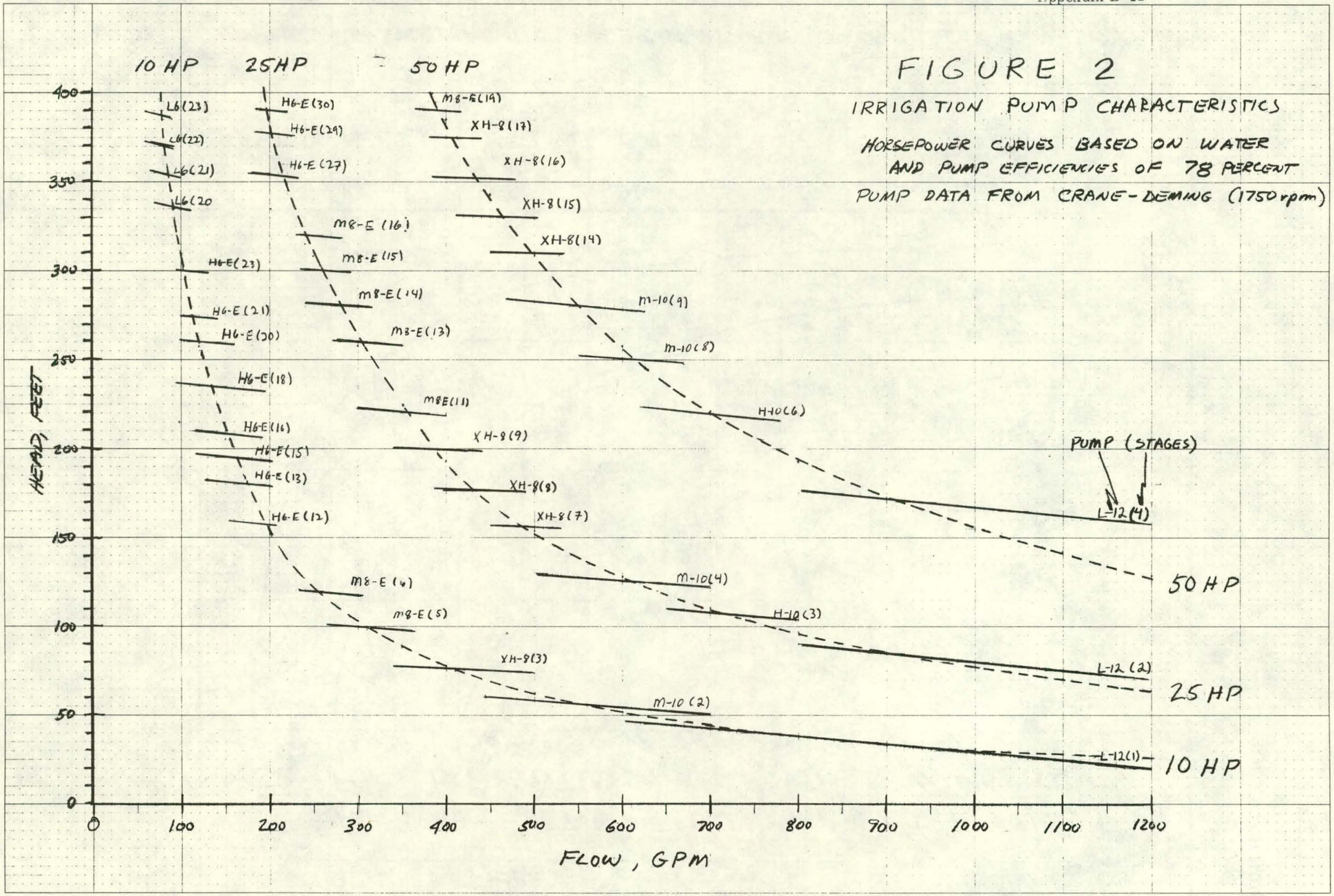


FIGURE 2
IRRIGATION PUMP CHARACTERISTICS
HORSEPOWER CURVES BASED ON WATER AND PUMP EFFICIENCIES OF 78 PERCENT
PUMP DATA FROM CRANE-DEMING (1750 rpm)
10 HP
25 HP
50 HP
PUMP (STAGES)
L6(23)
L6(22)
L6(21)
L6(20
H6-E(30)
H6-E(29)
H6-E(27)
H6-E(23)
H6-E(21)
H6-E(20)
H6-E(18)
H6-E(16)
H6-E(15)
H6-E(13)
H6-E(12)
M8-E(19)
M8-E(16)
M8-E(15)
M8-E(14)
M8-E(13)
M8E(11)
M8-E(4)
M8-E(5)
XH-8(17)
XH-8(16)
XH-8(15)
XH-8(14)
XH-8(9)
XH-8(8)
XH-8(7)
XH-8(3)
M-10(9)
M-10(8)
M-10(4)
M-10(2)
H-10(6)
H-10(3)
L-12(4)
L-12(2)
L-12(1)
HEAD, FEET
FLOW, GPM
0
50
100
150
200
250
300
350
400
0
100
200
300
400
500
600
700
800
900
1000
1100
1200

8/11/78 MD

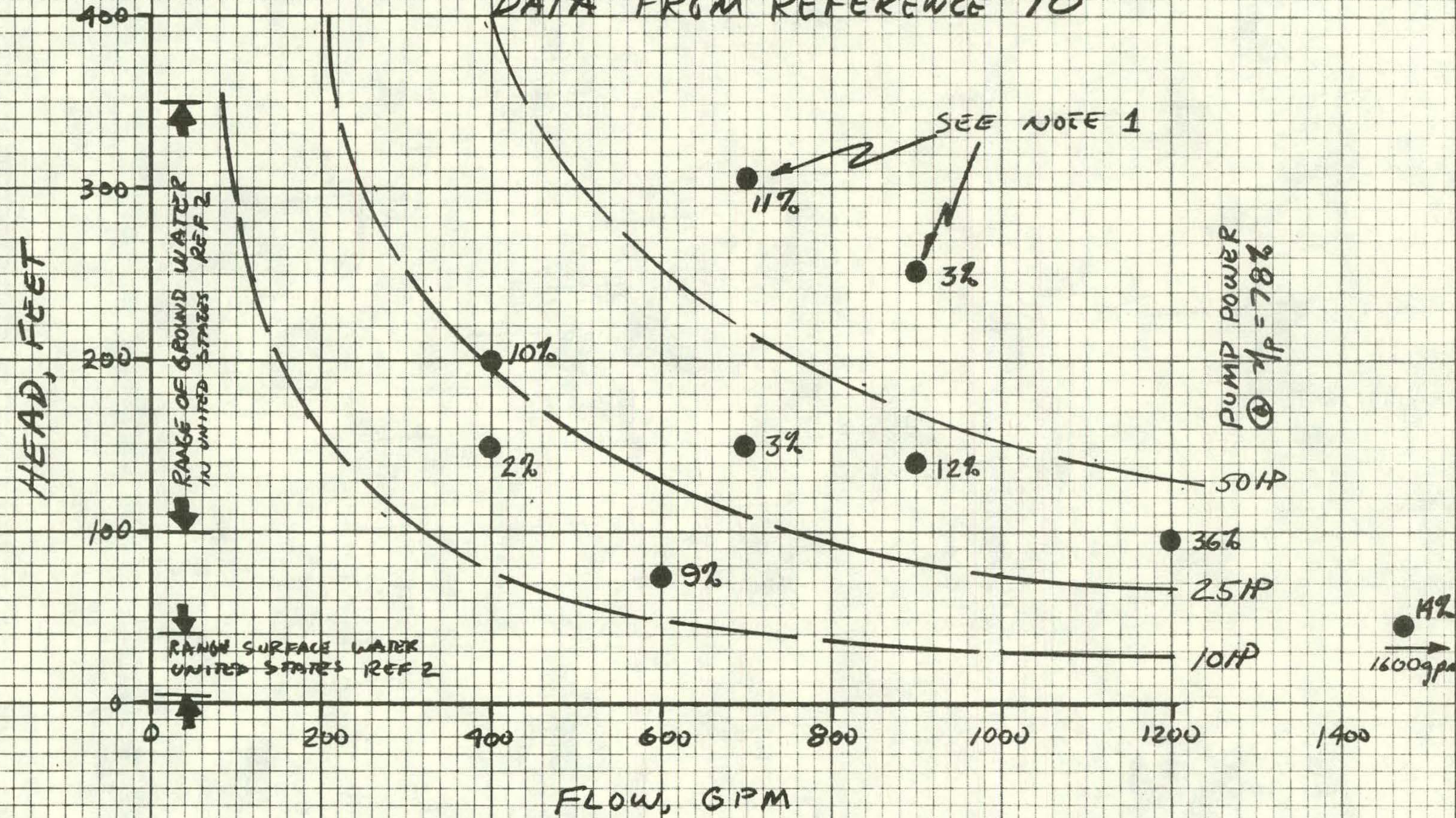
BARBER-NICHOLS ENGINEERING CO.

PROJECT DOE A-A
BY
DATE 10-18-78
SUBJECT

FIGURE 3

AVERAGE HEAD-FLOW CHARACTERISTICS FOR IRRIGATION PUMPS IN NEW MEXICO

DATA FROM REFERENCE 10



NOTE 1: NUMBERS INDICATE PERCENT OF FUEL COST ATTRIBUTED TO EACH HEAD-FLOW GROUP

Table 1

Regional Irrigated Area, Energy Used, and Average Lift (1974)
(Ref. 2)

	Pumped Irrigated Area (%)	Energy Used for On-Farm Pumping (%)			Lift (Ft.)	
		Ground	Surface	Total	Ground	Surface
Northern Plains	21	21	1	22	110	48
Southeast	6	2	0	2	197	13
Delta	8	3	0	3	84	13
Southern Plains	27	31	2	33	223	31
Mountain	16	24	1	25	252	15
Pacific	19	8	6	13	190	143
Other	3	2	0	2	-	-
Total	100	90	10	100		
Total U.S.	35.1×10^6 Acres	23.4×10^4 Billion Btu's	2.7×10^4 Billion Btu's	26×10^4 Billion Btu's		

Table 2

State Irrigated Area, Energy Used, and Average Lift (1974)
(Ref. 2)

	Pumped Irrigated Area (%)	Energy Used for On-Farm Pumping (%)			Lift (Ft.)	
		Ground	Surface	Total	Ground	Surface
Texas	25	27.9	1.8	30.7	200	40
Nebraska	14	11.8	.3	12.1	100	20
California	13	6.3	.1	6.4	110	10
Kansas	7	9.1	0	9.1	180	15
Arizona	3	8.4	0	8.4	350	0
New Mexico	2	8.0	0	8.0	350	5
Total 6 States	64	71.5	2.2	73.7		
Idaho	3	2.3	.2	2.5	275	0
Colorado	5	2.5	0	2.5	115	10
Washington	4	.8	4.0	4.8	250	250
Total 9 States	76	77.1	6.2	83.3		
Total U.S.	35.1×10^6 Acres	23.4×10^4	2.7×10^4	26×10^4		
		Billion Btu's				

Table 3

Farm Stationary Power Plants Estimated from Texas (Units/Acre) for States of: Arizona, California, Idaho, Nevada, New Mexico, Texas, Utah, and Washington (Ref. 6)

	Number of Units		
	Irrigation	Other	Total
Internal Combustion			
2-49 hp	22,000	14,000	36,000
50-199 hp	100,000	3,000	103,000
200 + hp	33,000	2,000	35,000
Electric Motors			
2-10 hp	63,000	29,000	92,000
11-99 hp	73,000	4,000	77,000
100 + hp	4,000	2,000	6,000
Total	295,000	54,000	349,000

Table 4

Distribution System Pressure Losses (Ref. 9)

System	Pressure (psi)	Assumed Water Application Eff.
Center pivot	80	.80
Side roll	60	.75
Big gun	110	.70
Solid set	50	.80
Hand move	60	.75
Mechanical move	100	.70
Drip	25	.95
Open ditch without return system	1	.50
Open ditch with return system	10.8	.60
Gated pipe without return system	7	.60
Gated pipe with return system	17.8	.70

Table 5

Variation of Energy Demand for Various Crops and Several Irrigation Means
(Ref. 9)

Region	Crop	Irrigation System	Peak Requirement: Peak Period, Month*	Peak Requirement: Water Appli. IN/PD**	Peak Requirement: Energy Demand w/Ground-water KWH/PD	Peak Requirement: Energy Demand w/Surface Supply KWH/PD	Annual Requirement: Annual Water Appli. IN/YR	Annual Requirement: Annual Energy Demand w/Ground-water KWH/YR	Annual Requirement: Annual Energy Demand w/Surface Supply KWH/YR
TEXAS	Sorghum	Ditch w/o PB	7-8	5	163	1	27	880	6
		Center pivot	7-8	4	225	95	20	1125	475
	Cotton	Ditch w/o PB	7-8	4	130	1	14	456	4
		Center pivot	7-8	2	112	48	14	785	335
	Wheat(forage)	Ditch w/o PB	10,12,4,5	4.5	147	1	27	882	6
		Center pivot	10,12,4,5	4	225	95	18.5	1040	443
	Wheat(no forage)	Ditch w/o PB	4-5	4.5	147	1	18	588	4
		Center pivot	4-5	3.5	197	84	12.5	703	300
	Corn	Ditch w/o PB	6-8	5	163	1	28	913	7
		Center pivot	6-8	4	225	95	20.5	1152	488
WASHINGTON	Hay	Ditch w/o PB	4-10	6.5	180	19	52	1439	151
		Side roll	4-9	4.5	204	92	35.5	1610	727
	Wheat	Ditch w/o PB	4-6	5	138	14	25	693	74
		Side roll	4-6	3	136	61	15	681	306
	Fruits-nuts-berries	Ditch w/o PB	5-10	6	166	17	57	1577	163
		Solid set	5-10	5	212	88	41	1735	721
	Vegetables	Ditch w/o PB	4-8	7	194	20	46	1275	134
		Hand move	4-8	5	226	102	31	1403	633

* Preirrigation peak ignored.

** PD = one-half month period

Appendix C-1

SOLAR POWERED RANKINE CYCLE IRRIGATION PUMP: COLLECTOR EVALUATION AND SELECTION PROCEDURE

Contract No. ET-78-C-03-1891

August 18, 1978

Prepared For: Department of Energy

Prepared By: Barber-Nichols Engr. Co.

6325 West 55th Avenue, Arvada, Colorado 80002

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Appendix C-3

1.0 INTRODUCTION

A form letter was sent out in February, 1978, to 19 solar collector manufacturers of a collector module that could be suitable for use in this application. Six manufacturers responded to this request. A copy of the request form letter and the manufacturers it was sent to is shown in the appendix.

Six companies responded with proposals to this request. The listing of these companies is provided on a separate sheet for DOE use only. Also, proprietary cost information provided is included in this separate sheet. All other data presented herein will be listed by company number only and relative cost values obtained during the calculation as described herein. Some pertinent parameters of the six proposed collectors are shown in Table I.

2.0 SELECTION APPROACH

To properly evaluate the collector concepts on the basis of system dollar cost per gallons of water pumped, it was decided to evaluate the cost of a 25 horsepower (peak) solar system. Horsepower related directly to the gallons of water pumped in that the product of the flow rate and the head determines the horsepower and, consequently, the gallons of water pumped by a 25 horsepower system will vary depending on the specific application. However, horsepower and gallons of water pumped will relate on a 1:1 basis for a given local. Therefore, it was decided to evaluate all systems on the basis of a system which would produce 25 horsepower during a total solar intensity of 300 Btu/hr-ft^2. Ultimately, the systems were compared on the basis of the total system installed cost per horsepower hour during a year of operation. This calculation was made as described below.

Initial calculations were made based on the precedure presented in reference 1, where the different basic collector types were represented by a single curve as shown in Figure 1, obtained from reference 1. Although all concentrators do not operate on the same operating line, it has been found that this figure does represent collectors to the first approximation. When the Rankine cycle engine efficiency as a function of temperature is combined with the collector efficiency the overall solar conversion efficiency can be obtained and optimized as a function of collector output temperature as shown in Figure 2, reproduced from reference 1. These curves were used to determine the optimum operating efficiency of the six collector systems proposed from this program. Although some differences did occur between the curves of Figure 1 and the proposed systems, it was felt that these differences were minor in the overall evaluation and may not prove to be actual, consequently, all systems were evaluated using Figure 2.

Table II itemizes many of the parameters presented by the proposers and also shows in columns 1 and 2, the optimum operating temperature and the system efficiency for five of the proposed systems. The sixth proposal

TABLE I

Mfg.	Type	Rec. Tube	Config.	Module Size	Rec. Heat Flux (Btu/hr-ft^2)	Comments
Del Mfg.	Trough	Fixed	Tracking E-W	$2' \times 8' = 16$ ft^2	21:1-3800	Glass mirror refl. - many parts-small module-approx. 200 ft^2 made-orders for 40,000 ft^2 pending-testing now at Sandia
Thermo Kinetics	Trough	Moving	Tracking E-W	$4' \times 20' = 80$ ft^2	31:1-5500	Bellow flex lines-heavy 320 lb/unit -hydr. drive
SunTech	Slats	Fixed	Tracking N-S	$10' \times 20' = 200$ ft^2	Rec. not spec.	Rec. high off ground-many parts-glass mirror-questionable if boiling concept will work
Alpha Solar	Trough	Unknown	Tracking E-W	6 col$\approx$108 ft^2	Rec. not spec.	Six collector package
Maritime Dynamics	CPC	Fixed	Fixed	$4' \times 8' \approx 26$ ft^2	Low	Non-tracking - $\pm$ 25^o acceptance angle-acrylic cover-100 ft^2 made-has boiled R-113 in collector
Energy Design	CPC	Fixed	Fixed	$4' \times 8.7' \approx 28$ ft^2	Low	U-tubes - not acceptable for boiling in collector concept

FIGURE 1
ESTIMATED PERFORMANCE OF SEVERAL TYPES OF SOLAR COLLECTORS (from Ref. 1)

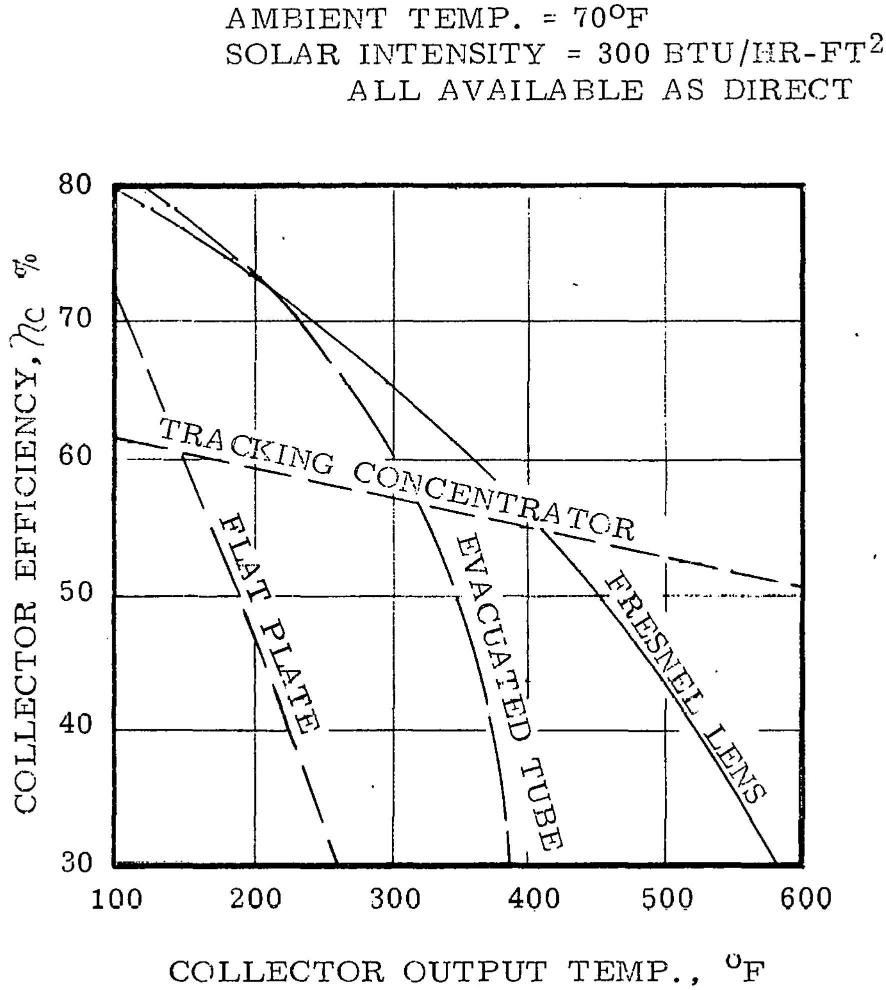


FIGURE 2
ESTIMATED SOLAR CONVERSION SYSTEM EFFICIENCY AS A FUNCTION OF COLLECTOR TEMPERATURE (from Ref. 1)

ASSUMPTIONS:
1) SOLAR INTENSITY-300 BTU/HR-FT^2 90% DIRECT
2) MAX. CYCLE TEMP. = 95% COLLECTOR TEMP.
3) COLLECTOR EFF. FROM FIGURE 1
4) RANKINE CYCLE WITH REGENERATION
5) INDIRECT COMPONENT LOST TO FRESNEL AND CONCENTRATOR

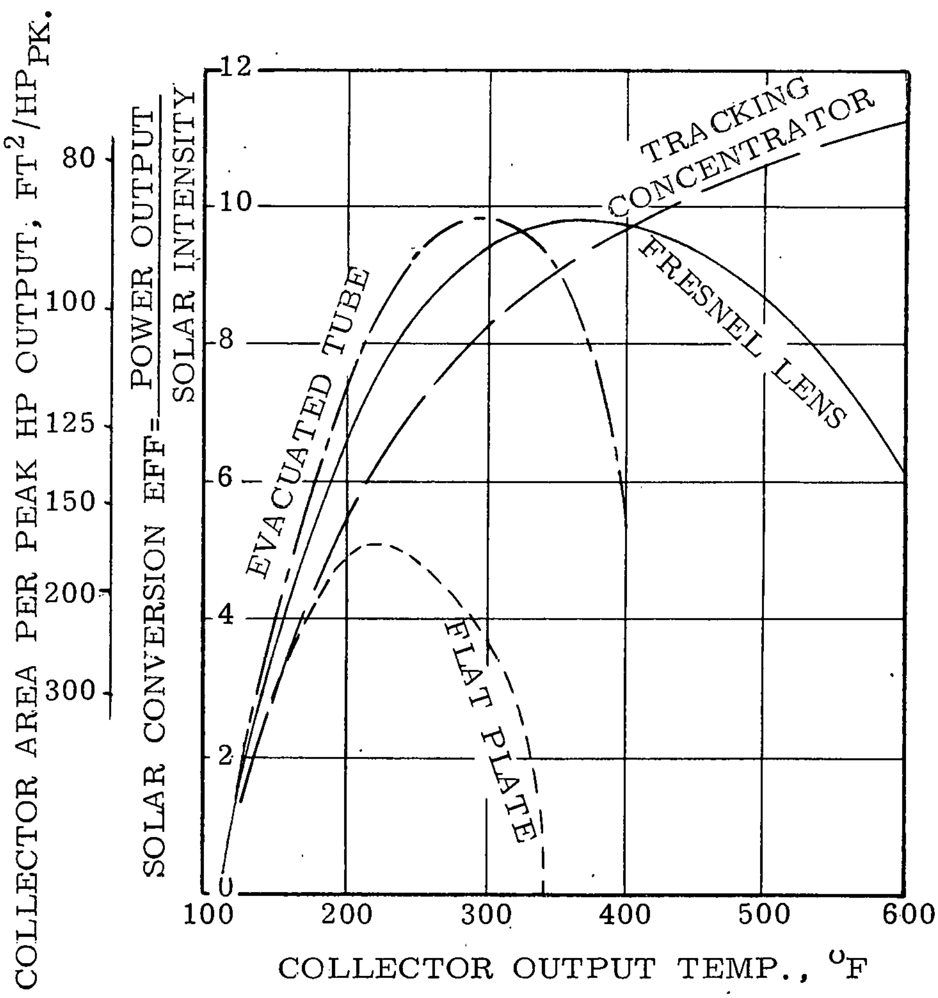


TABLE II

25 HP ENGINE PEAK

Company		Opt. T_o	η_s	Mult. for Orient.	Mult. for Direct.	ft^2/hp_p	ft^2	$\$_c/hp_p$	Rel. Cost
				Solar Collector				Installed Costs: Engine	
1	Now	400	10%	1.5	.9	94	2350	2000	1.0
	Prod.	400	10%	1.5	.9	94	2350	480	.47
2	Now	400	10%	1.5	.9	94	2350	2000	1.09
	Prod.	400	10%	1.5	.9	94	2350	480	.61
3	Now	400	10%	1.4	.9	94	2350	2000	1.25
	Prod.	400	10%	1.4	.9	94	2350	480	.64
4	Now	400	10%	1.5	.9	94	2350	2000	1.44
	Prod.	400	10%	1.5	.9	94	2350	480	.40
5	Now	220	5.5%	1.0	1.0	154	3860	2000	1.97
	Prod.	220	5.5%	1.0	1.0	154	3860	600	.89

was not considered to be acceptable because of the U-tube arrangement in the collector which was felt to be unacceptable for a boiling in the collector concept. The values in Table I that are missing are provided for DOE information in the confidential sheet included in the attachment hereto.

It was arbitrarily assumed that the collectors which require high concentration ratios can only make use of the direct component of the solar insulation and it was assumed that the direct component was 90% of the total insulation. Company five is the only collector concept presented that could utilize the total solar insulation. This multiple for direct, column 8, along with the cycle efficiency, column 2, allows one to calculate area required in square feet for peak horsepower in column 9 and the total collector size for a 25 horsepower system shown in column 10 of Table II. Utilizing the square feet figures, the dollars per horsepower peak of collector, column 11, was evaluated for the vendor's estimate of production cost and the quotation for the small collector field presently quoted upon shown as the now value in Table II.

The Rankine engine cost was evaluated similarly to that shown in reference 1. However, reference 1 assumes a 50% installation factor for the engine. It is the author's opinion that this number is excessively high for irrigation type installations when considering that the engine will be a packaged unit only requiring a concrete pad and a small building for protection from the weather. Therefore, it was assumed that the installation cost of the engine was 20% for irrigation applications. Since the engines in reference 1 have components not required in the boiling-in-the-collector concept, namely the boiler, preheater, and certain controls, the cost of the boiling-in-the-collector systems were found to be approximately 25% less when evaluating the component cost necessary in making up the system. Consequently, the data shown in Figure 3 was obtained for a 25 horsepower system as a function of maximum cycle temperature, where the dashed line is for conventional Rankine engines and the solid line is for the boiling-in-the-collector Rankine engines. The installed engine cost is a function of cycle temperature and is higher for a 200°F cycle temperature than the 400 or 500°F cycle temperature manily because the heat loads and, hence, heat exchanger size is substantially higher because of the lower engine efficiency at the lower temperature. The engine costs shown in Figure 3 were used for the calculations summarized in Table II and are indicated in column 12 for the production units. The "now" unit cost of $2000/horsepower was estimated as a reasonable value for one-of-a-kind units at the present time. By adding the engine and collector costs the total dollars per horsepower peak were obtained and shown in column 13.

Orienting collector systems will collect more solar energy because of the ability to collect early and late in the day. The data in reference 2 shown in Figure 4 was utilized to provide a multiple for orientation as shown in column 7 of Table II. In this case, manufacturers one, two, and four utilized a N-S axis for the collector oritneing east and west, and were given a 1.5 multiple. Manufacturer five utilized a flat plate collector, therefore obtaining a 1.0 multiple, and manufacturer three utilized an E-W axis orienting north and south, therefore resulting in a 1.4 multiple. This multiple relates to

FIGURE 3

ESTIMATED RANKINE ENGINE INSTALLED COSTS FOR PRODUCTION IRRIGATION SYSTEMS
(from Ref. 1)

25 Horsepower
1000 Units per year
Installation cost = 20% of equipment cost

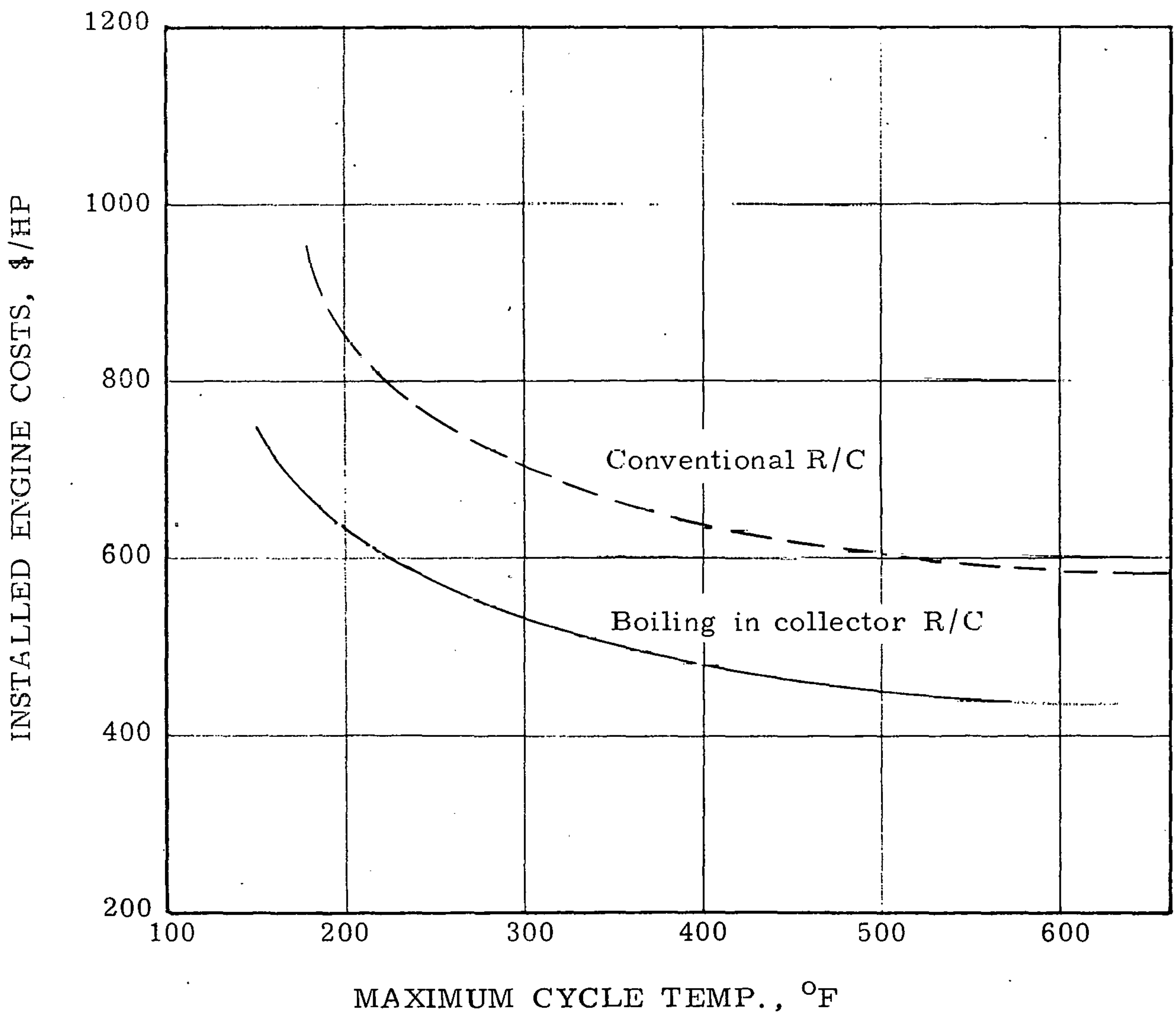
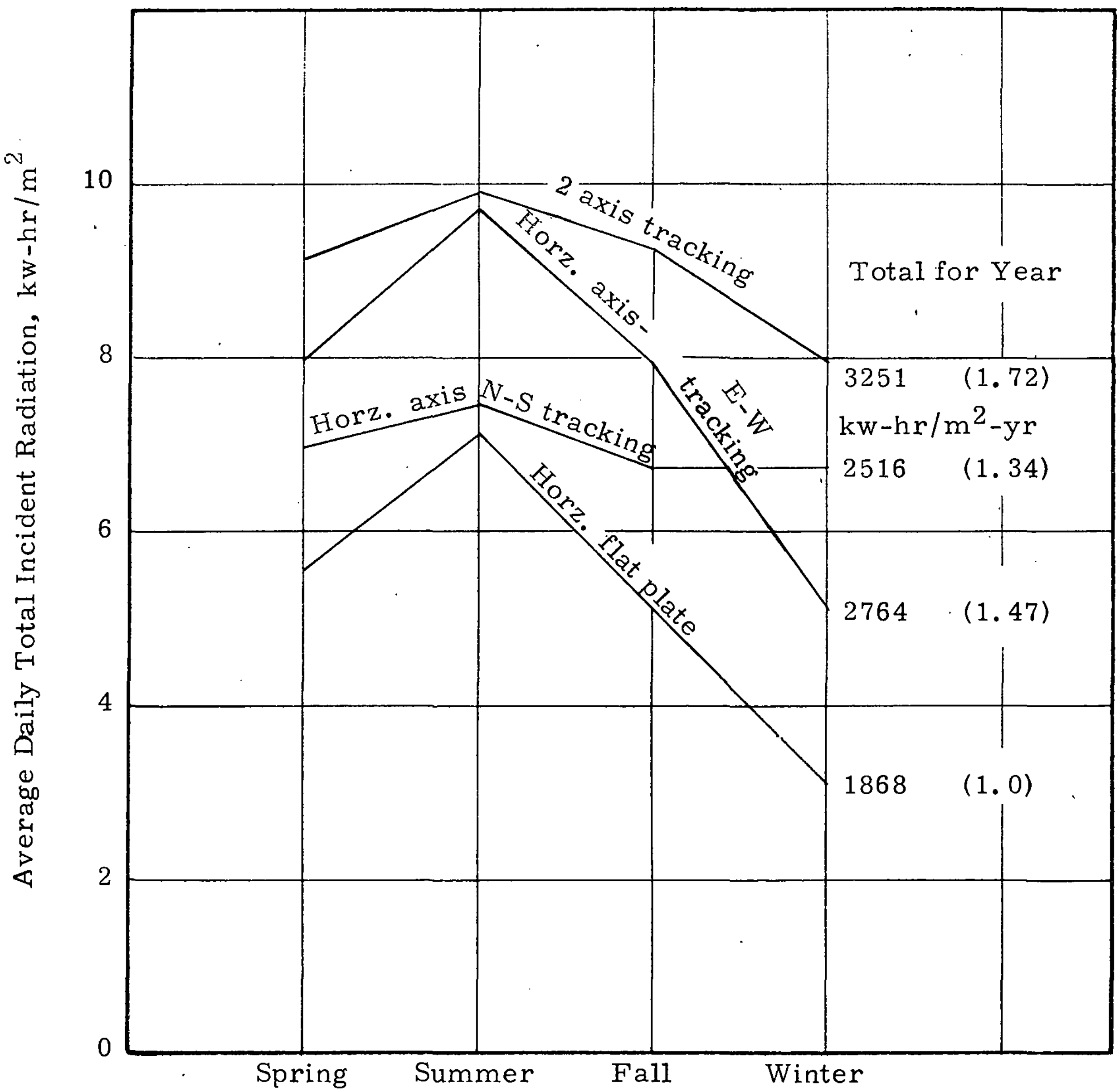


FIGURE 4

CLEAR DAY AVERAGE DAILY SOLAR RADIATION AVAILABILITIES - ALBUQUERQUE, REF. 2



the hours of operation of the particular collector type, therefore, when obtaining the system costs per horsepower hour, the system cost per horsepower was divided by the multiple for orientation. This absolute value is meaningless, however, as a relative cost basis it indicates the total yearly relative cost of the different approaches. The relative cost shown in column 15 was evaluated based on the absolute value of manufacturer one present calculated system cost for convenience. It can be seen here that the current cost of the system varies from 1 to 2, and the production cost varies from .4 to .9 as graphically shown in Figure 5.

It is concluded from Figure 4 that the orientating high temperature systems have essentially the same current cost and relatively the same production cost regardless of the specific type and manufacturer. The non-orienting concentrator has substantially higher current and production costs on the basis of horsepower hour because of the lower system efficiency and lower collection time, although the absolute system cost on a dollar per peak horsepower is not substantially different. The non-orienting system will certainly have a higher reliability factor than the orienting systems which are more complex and more prone to failure. However, until time proves what the reliability of these systems are, it is felt that the orienting concentrating collector should be selected for the present study. Additionally, contractor one and two were selected to supply approximately 1/2 each (150 sq. ft.) of the collector field for this study.

Two collectors were selected even though the costs exceed the budget in this fixed price program because it was felt that contractor one, Del Manufacturing Co., has a superior gloss mirror reflector and a small, 2 feet wide by 8 feet long, collector while contractor two, Solar Kinetics, had the larger 4 feet wide by 20 feet long collector which should have lower installation and production costs, and represents a number of other manufacturer approaches (i.e. Accurex, Hexcel, etc.).

3.0 PERFORMANCE OF SELECTED COLLECTORS AND ENGINE

When the vendor data of the selected collectors were plotted there was a difference as shown in Figure 6. Vendor one has lower efficiency at the higher temperatures primarily because of the lower concentration ratio which results in higher receiver losses. Also shown in Figure 6 is the Rankine cycle engine efficiency as a function of temperature. The engine efficiency was calculated for a condensing temperature of 95°F which is generally high for normal irrigation applications, however, this engine efficiency also assumes a turbine efficiency of 80%, which is also high, for a 25 horsepower application. Therefore, the overall net result is that the data shown in Figure 6 is reasonable for this application. When combining the engine and collector efficiency, the overall solar conversion efficiency is shown in Figure 7. For a direct solar insolation level of 300 Btu/hr-ft^2 an engine efficiency between 11% and 12% would be expected at 400°F, and at a solar insolation of 200 Btu/hr-ft^2, the efficiency (peak) is 9% for manufacturer one and 10.5% for manufacturer two. Therefore, over normal operating conditions for concentrating collectors, high efficiency would be expected for this system. However, additional analysis

FIGURE 5

RELATIVE COST OF 25 HP SOLAR IRRIGATION
SYSTEM USING VARIOUS TYPES OF COLLECTORS

Production cost for 1 x 10^6 ft^2/yr.

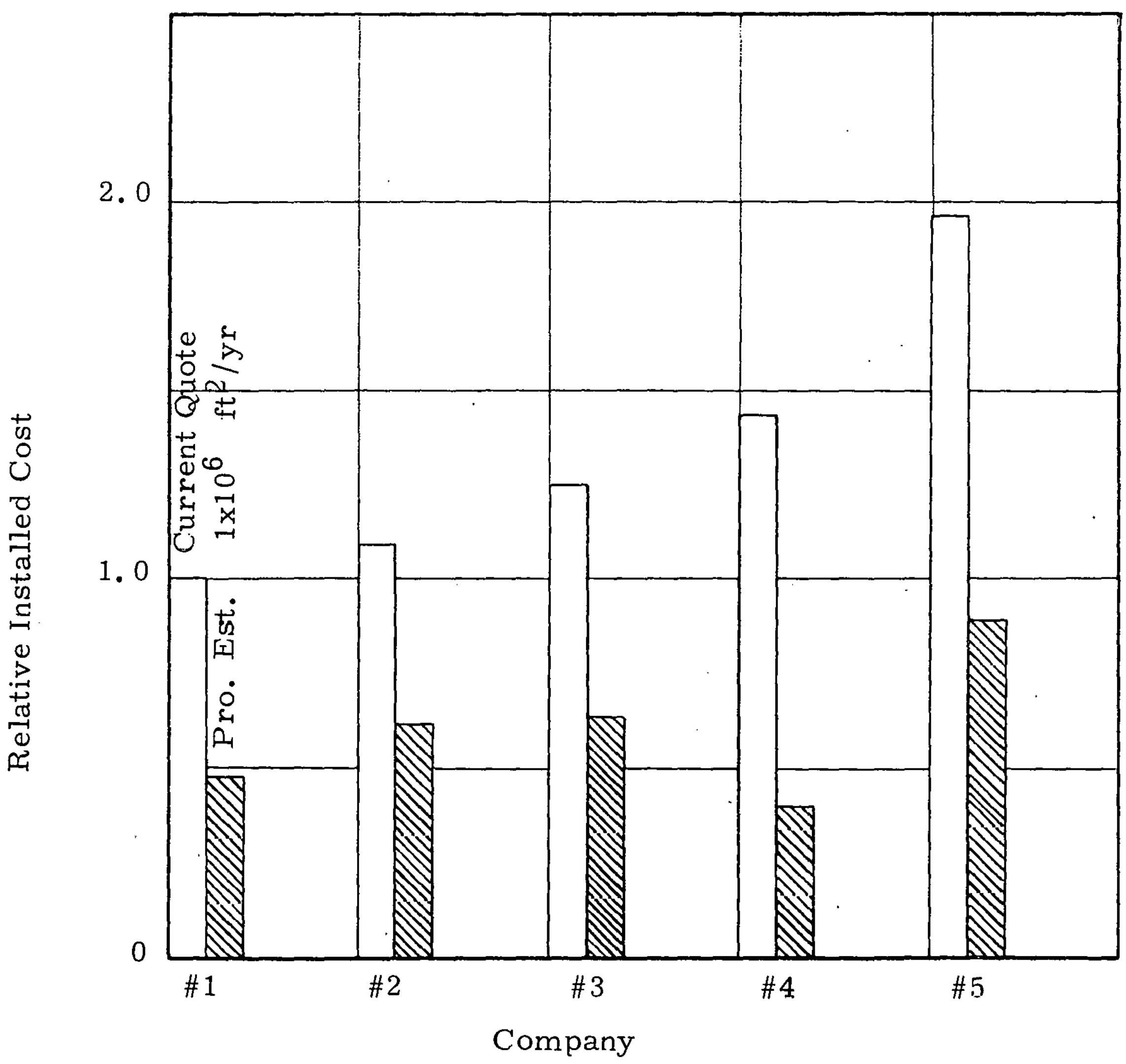


FIGURE 6

ESTIMATED RANKINE ENGINE AND COLLECTOR EFFICIENCY

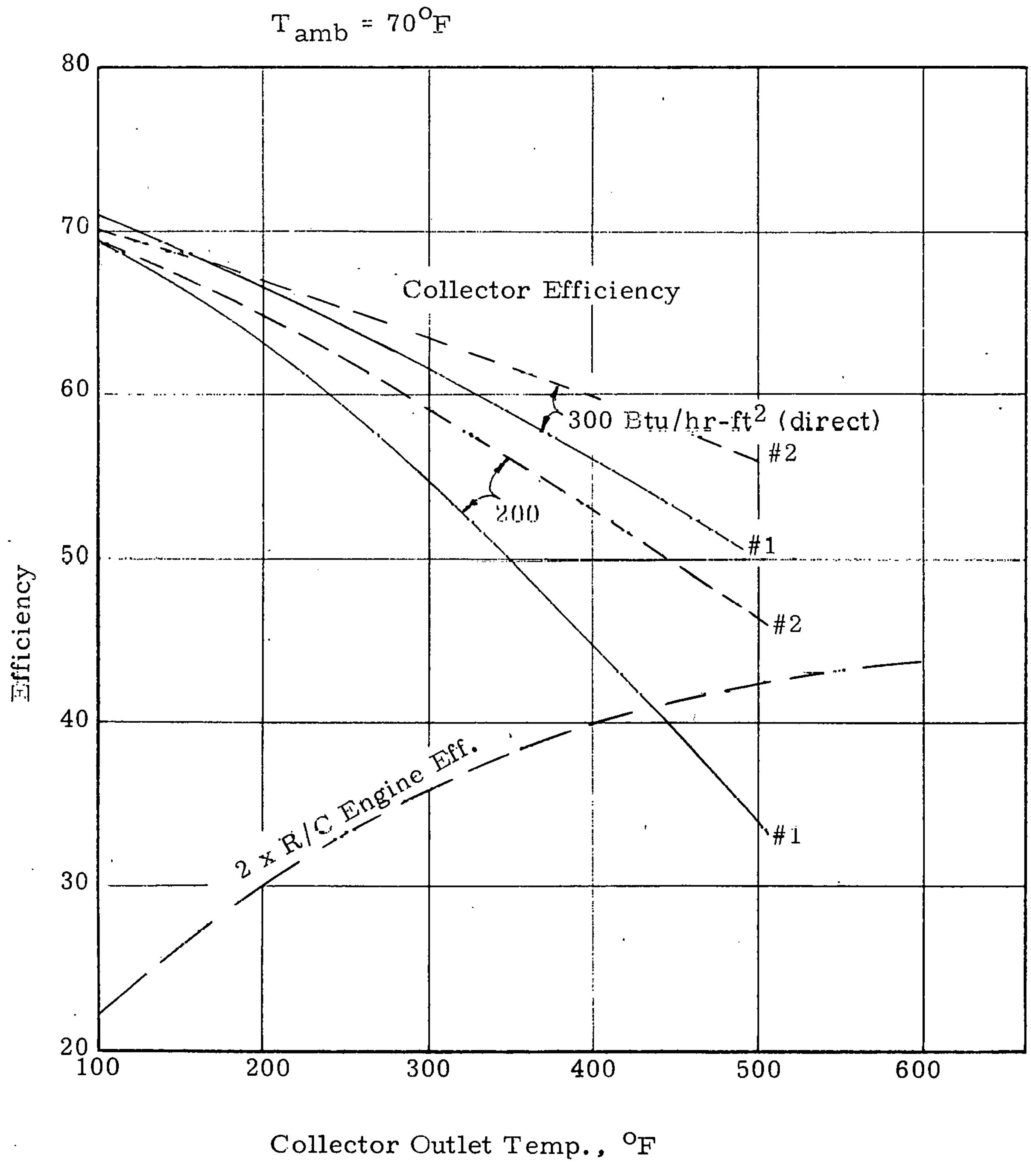
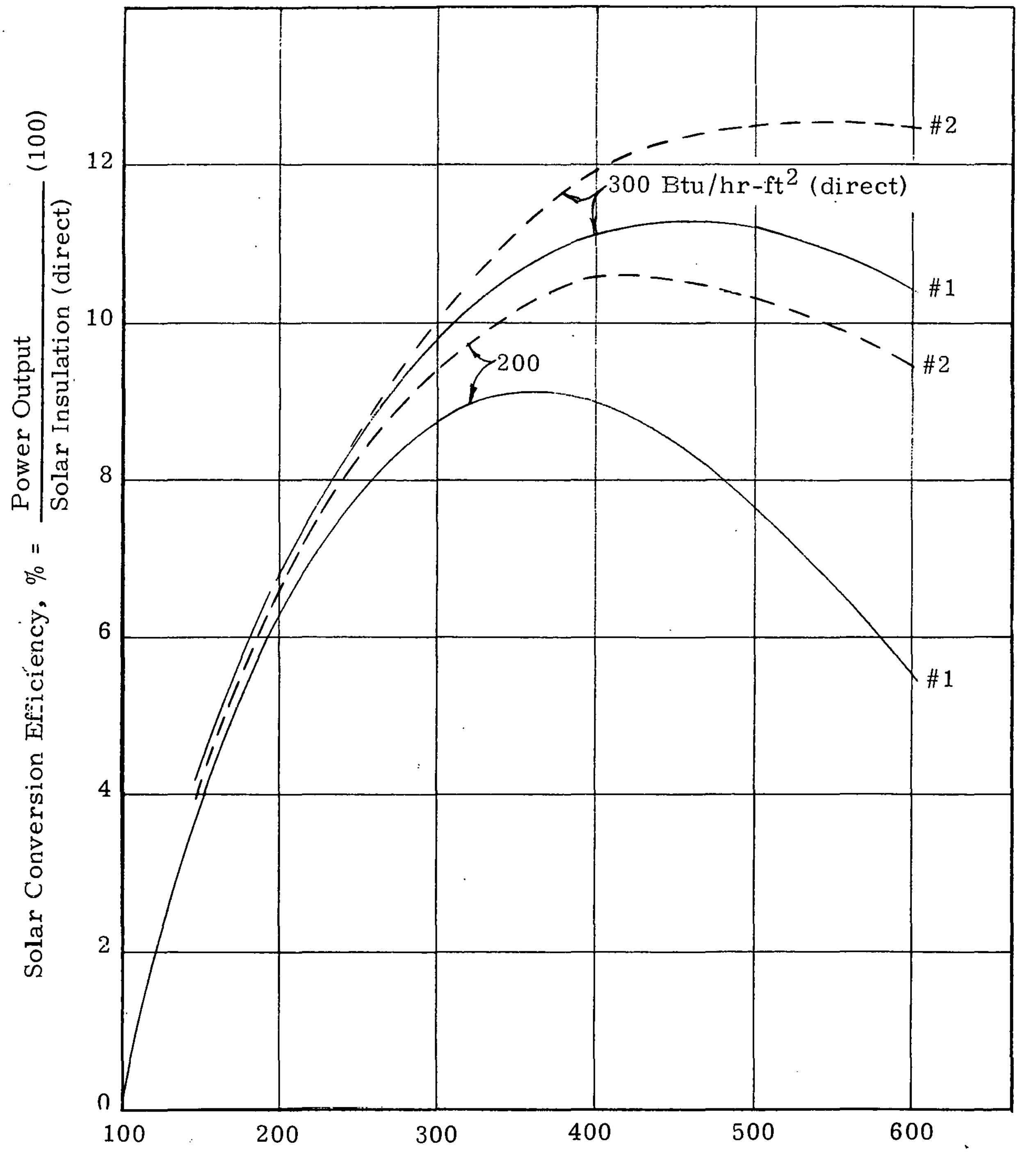


FIGURE 7

ESTIMATED SOLAR CONVERSION EFFICIENCY

25 Horsepower



combining the engine off-design characteristic must be included before determining the actual efficiency of the system as a function of solar insolation. These calculations will be made later in the program.

Optimum efficiencies occur at a temperature greater than 400°F for high solar insolations. However, refrigerants are not suitable working fluids for temperatures above 400°F, and since the increase in efficiency for higher temperature operation is small, it is recommended that the system be designed for a maximum temperature of 400°F so that R-113 can be utilized.

4.0 REFERENCES

1) Barber, Robert E., "Current Costs of Solar Powered Organic Rankine Engines", Solar Energy, Volume 20, No. I, 1978, Pages 1-6.

2) Boes, Eldon C., "Solar Radiation Availability to Various Collector Geometries: A Preliminary Study", Sandia report SAND76-0009, February, 1976.

APPENDIX I

Maritime Dynamics, Inc.
1169 East Ash Avenue
Fullerton, CA 92631

Owens-Illinois, Inc.
P.O. Box 1035
Toledo, OH 43666

Environmental Designs
P.O. Box 12408
Memphis, TN 38112

Trimlite
25 Holden Street
Providence, RI 02908

Falbel Energy Systems Corp.
472 Westover Road
Stamford, CN 06902

Northrup, Inc.
302 Nichols Drive
Hutchins, TX 75141

Alpha Solarco
Suite 2230
1014 Vine Street
Cincinnati, OH 45202

Energy Applications, Inc.
830 Margie Drive
Titusville, FL 32780

Acurex Aerotherm
485 Clyde Avenue
Mountain View, CA 94042

Honeywell, Inc.
Energy Resources Center
2600 Rdigway Parkway
Minneapolis, MN 55413

Del Manufacturing Co.
905 Monterey Pass Road
Monterey Park, CA 91754

Scientific-Atlanta, Inc.
3845 Pleasantdale Road
Atlanta, GA 30340

Chamberlain Manufacturing Co.
845 Larch Avenue
Elmhurst, IL 60126

Hexcel
11711 Dublin Boulevard
Dublin, CA 94566

Sheldahl
Advanced Products Division
Northfield, MN 55057

KTA Corp.
12300 Washington Avenue
Rockville, MD 20852

Entropy, Ltd.
5735 Arapahoe Avenue
Boulder, CO 80303

Solar Kinetics, Inc.
147 Parkhouse Street
Dallas, TX 75207

General Electric Co.
P.O. Box 13601
Philadelphia, PA 19101

APPENDIX II

February 9, 1978

Barber-Nichols Engineering Company has a contract to do development work on an improved solar power irrigation system. This system will utilize Refrigerant 113 in the collector receivers and the Rankine power loop. The goal of the project is to develop a lower cost, higher performance system by utilization of a unique approach. The approach is to directly boil the Rankine working fluid in the collector thereby eliminating the need for a boiler. Other system advantages occur from this approach but do not affect the collector. The present phase of the program is to prove the concept of the collector and the engine by test. The next phase will be to field test a prototype unit.

In this phase a collector module of approximately 300 ft^2 will be tested. Both fixed and orienting collectors will be considered. The selection of the collector will ultimately be made based on the total system cost per horsepower output.

In order to increase the output of the pumping system Barber-Nichols feels that the collectors should be a north-south mount with east-west orientation. The north-south tilt angle must be great enough to provide adequate gravity head of liquid (see the attached sketch) to provide circulation of the refrigerant in the receiver tube during maximum boiling rates.

The collector field arrangement is shown in the attached sketch. This configuration is probably not that of your present collector; however, this approach has many system advantages. It is suggested that the collector vendor be responsible for the reflector, receiver, orientation system, vapor and liquid header, and mechanical support structure. Barber-Nichols will be responsible for the separator tank, supply of the refrigerant to the liquid header, control center, and all Rankine cycle components. Barber-Nichols can be of assistance to you in the design of the receiver if necessary. The orientation system should have provision to be defocused by the control center in the event of malfunction or overtemperature. The connections between the receiver and the top and bottom headers should have rigid attachments and the plumbing must be leak tight to prevent refrigerant leakage.

Barber-Nichols will issue a purchase order for a collector module of approximately 300 ft^2 in April for delivery two months later (June) based on responses to this letter. This unit will be tested for approximately four months while the Rankine engine is also being checked out on a laboratory test setup. The results of both of these tests will be combined to result in a final report, near year end, summarizing this project and presenting the cost and schedule for Phase II which will be to build and test one or more field test units.

To evaluate your collector we will need, as a minimum:

a) The collector performance as a function of solar insolation and receiver liquid temperature (200°F to 400°F of interest);

b) The collector and receiver tube geometry so that receiver tube heat flux and pressure drop can be determined;

c) Your willingness and capability to meet the schedule above;

d) Cost for the test module and projected cost in quantities of 10,000, 100,000, and 1,000,000 ft^2/year;

e) Projected installation cost of a 2,000 ft^2 collector field; and,

f) Any additional information you feel will be of value to us.

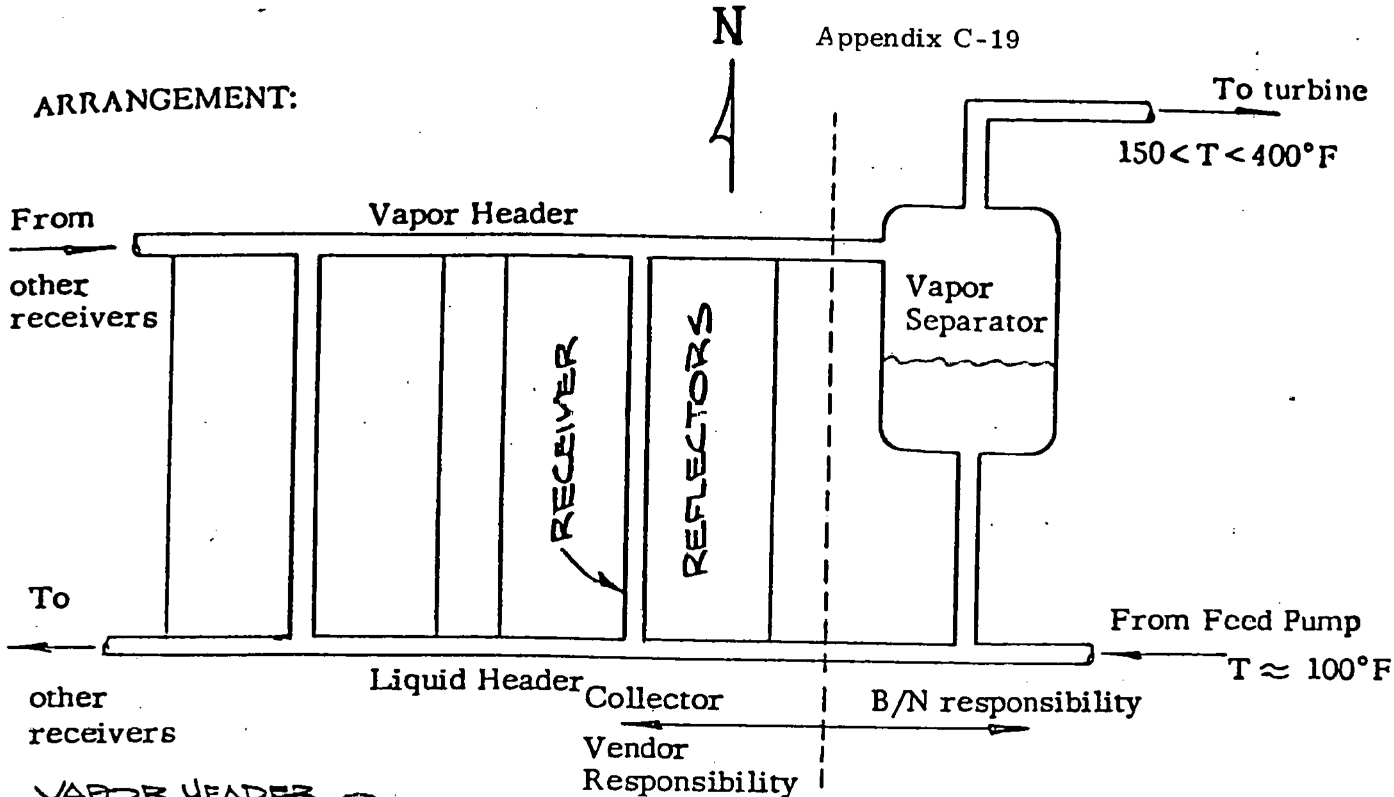
If you wish to be considered in this procurement, please respond to me by the end of February. If you have any questions, feel free to contact me, or Daryl Prigmore. I look forward to hearing from you.

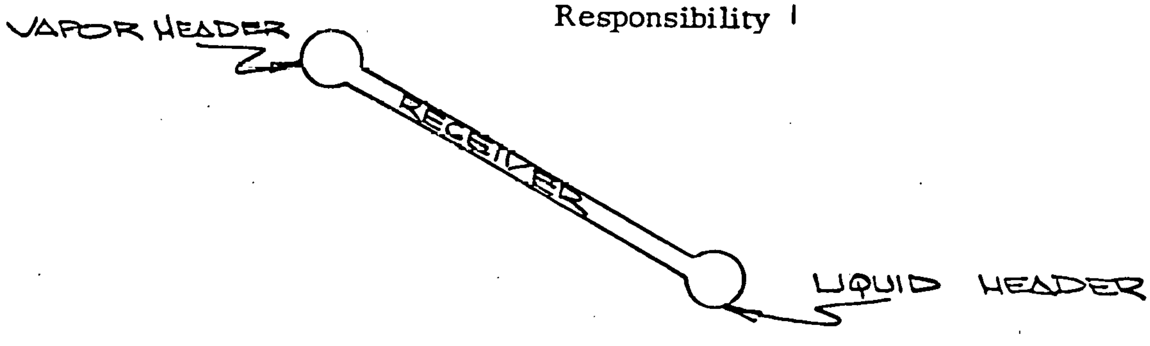
Sincerely,

Robert E. Barber

jm

ARRANGEMENT:





SIDE VIEW

RECEIVER REQUIREMENTS:

1) Pressure drop less than one-half tilt height.
2) Maximum allowable local heat flux on receiver inside surface 4000 Btu/hr-ft^2.
3) Maximum operating pressure 425 psia (@ 400°F). Proof test at 675 psi. Maximum overpressure 450 psia (also @ 400°F).
4) No hose couplings or rotating couplings.
5) No leaks in or out. (Welded pipe preferred, steel ok-no copper.)

ERROR ANALYSIS

Collector Efficiency

$$\eta_c = \frac{\dot{m}\ \text{R-113}\ (h\text{sys out} - h\ \text{sys in}) + K_1\ (T_{pipe} - T_{ambient})}{I * A}$$

Where $\dot{m}$ R-113 is refrigerant mass flow measured by sharp edged orifice.

$\dot{m} = 359\ K_2\ d^2\ Fa\ \sqrt{Hw\ \rho} = 373$ #/hr

$K_2 = .60 \pm .03$

$d = .324 \pm .003$

$Fa = 1 \pm .005$

$Hw = 40 \pm 3$

$\rho = 6.8 \pm 1.0$

$$\Delta\dot{m} = [\ (621.5 * .03)^2 + (2302 * .003)^2 + (373 * .002)^2 + (4.66 * 3)^2 + (27.4 * 1)^2]^{1/2}$$

$= 36.6$ #/hr

$\therefore \dot{m} = 373 \pm 36.6$ #/hr.

h sys out $= 120 \pm 2$ Btu/#

h sys in $= 26 \pm 3$ Btu/#

K_1 = Thermal conductance of insulation $= 12 \pm 4$

$T_{pipe} = 300^{o} \pm 100^{o}F$

$T_{amb} = 80^{o} \pm 40^{o}F$

$I = 300 \frac{Btu}{hrft^2} \pm 30$

$A = 288\ ft^2 \pm 5$

$\eta_c = 43.6\% \pm \Delta\eta_c$

$$\Delta\eta_c = [(.0011 * 36.6)^2 + (.0043 * 2)^2 + (.0043 * 3)^2 + (.0025 * 4)^2 + (.00013 * 100)^2 + (.00013 * 40)^2 + (-.00145 * 30)^2 + (-.00145 * 5)^2\ ]^{1/2} = .064$$

Therefore:

$\eta_c = 43.6 \pm 6.4\%$

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