# SYSTEM PERFORMANCE FOR A DISTRIBUTED FIELD OF LARGE POINT-FOCUSING COLLECTORS PRODUCING STEAM

**W. E. Rogers**
Power Kinetics, Inc.
Troy, New York

### Abstract

Power Kinetics has completed the testing of a 300 $m^2$ point-focusing Square Dish solar collector which was used to produce the steam to power a reciprocating steam engine. This paper uses data from this collector and engine module to project the performances of various size fields of these collectors with their piping losses. Comparisons are made to the reported performance of other systems and the results of other studies of field piping of solar collector arrays.

Losses through the insulation on the field piping carrying steam to simple Rankine cycle plants up to 50 $MW_e$ size are less than 4% at temperatures up to 550°C. Overall thermal performance in the same plants are shown to be greater than 80%. By using new strategies with today's large dishes, central engine dish systems offer viable opportunities to demonstrate the potential of solar thermal technology.

## INTRODUCTION

Distributed solar thermal collectors require a network of insulated pipes to carry heat from their receivers to the process where it is utilized. To quote James Leonard: "The performance of the thermal transport system is a higher leverage parameter than cost.... Any improvement in the efficiency of delivering energy to the point of use will result in a proportional decrease in the size of the solar collector field needed to achieve a given annual production-capacity." He goes on to say, "The heat capacity of the system is an energy tax that must be paid on each system start, and one which does not pay back in full at the end of the day" [1]. By considering both the collectors and the piping field together, the cost to deliver energy from the sun can be minimized.

This study addresses the use of water/steam as the heat transfer medium for solar collector fields. Performance curves are developed and the thermal response of the collectors and piping to a cold start in the morning and cloud transients are addressed.

## PERFORMANCE: DISTRIBUTED COLLECTOR FIELDS

The first major installation of dish solar collectors (114) was at Shenandoah, Georgia. Overall power from the field was measured at 2040 kW which is 52.6% of the available insolation and includes a field loss of 4.6% (@400°C) [2,3]. Other tests run on days with less available solar, gave field performances of between 42 and 46% [2,4]. The Australian National University designed, built, and installed at White Cliffs, New South Wales, a fourteen dish field of parabolic dishes which produce steam. This field delivers 173 kW peak at 360°C at an overall efficiency of 62%. Field losses from the bases of the collectors to the Power Module are 4% [5]. Table I lists some parameters of these projects. LaJet has installed a large field of 700 dishes. The steam producing collectors were divided into two groups, the periphery of the field produced saturated steam while the ones closest to the steam turbine were used to superheat the steam. Performance has not been published. It is known that considerable damage has been done by water freezing in pipes.

Many more projects involving parabolic trough collectors have been installed and their characteristics reported. Generally, most overall performances before Luz were in the 25 to 35% range with field piping losses around 8%, see Table I [1,6,7,8]. It should be noted that if a heat transfer fluid is used and steam is the end working fluid of the process, an additional thermal loss (3%, [4]) and a larger availability loss (because of the temperature degradation) are incurred.

| PROJECT | m2 | Temp. °C | System Performance | Field Piping Losses | Fluid |
|---|---|---|---|---|---|
| Dish: | | | | | |
| Shenandoah [1,2,3,4] | 4,387 | 400 | .42 - .53 | 4.6% | Syltherm |
| White Cliffs [5] | 277 | 360 | .50 - .62 | 4-7% | Steam |
| Trough: | | | | | |
| Coolidge [1,6] | 2,137 | 270 | .26 - .35 | 8% | Caloria |
| Dow [7] | 929 | 210 | .25 - .33 | 7% | Dowtherm |
| Luz-Segs III (est.) [8] | ~204K | 327 | .42 - .50 | 6-8% | VP-1 |
| Luz-Segs VIII (est.) | ~500K | 371 | .42 - .50 | 6-8% | VP-1 |

TABLE I: THERMAL PERFORMANCE OF DISTRIBUTED CONCENTRATING SOLAR COLLECTOR INSTALLATIONS





Dap: rogersandy

Estimates of the energy required to heat up the heat transfer loop containing a fluid in Shenandoah represents 17% of the available solar energy on a typical day, [2]. For SEGS III, it takes about an hour to bring the heat transfer loop with its fluid inventory to the operating temperature and deliver steam to the turbine,[8]. On this larger plant, located in an area with better solar availability, the proportion of the daily collected energy stored in the heat capacity of the fluid loop is smaller.

Various models for optimizing field piping systems have been partially developed. Jane Diggs compared the Larsen/Akau code with the ETRANS code for minimizing costs. For a sample system using Syltherm, piping losses were 7.7 and 7.5%, respectively, with the distribution of losses between daytime, overnight and pumping: 81, 16, and 3% for the first code, and 62, 30, and 8% for the second [9]. A comparison of the performance and economics of thermochemical and some sensible energy transport media was done but did not include water/steam. The study did consider an all steam system but the high cost of a compressor and its parasitic power and the sizes of the pipes made this approach uneconomical [10].

A very encouraging study demonstrates the potential for distributed collector solar systems to penetrate the industrial process heat market by illustrating the potential to achieve low-cost thermal transport piping networks. Semi-automated field installation of network components is recommended, with installed costs 40% less than conventional methods. Estimates of thermal transport losses range from 2% at 100°C to 9% at 500°C in 30 to 100 MWth plants [11]. Another reference quantifies the heat losses in a carefully instrumented test loop. Calculations are compared to test results and shown to be very comparable and care should be taken to eliminate pipe hangers and their thermal problems. It is also shown that the heat loss through an insulated control valve is equivalent to 4 meters of insulated pipe [12].

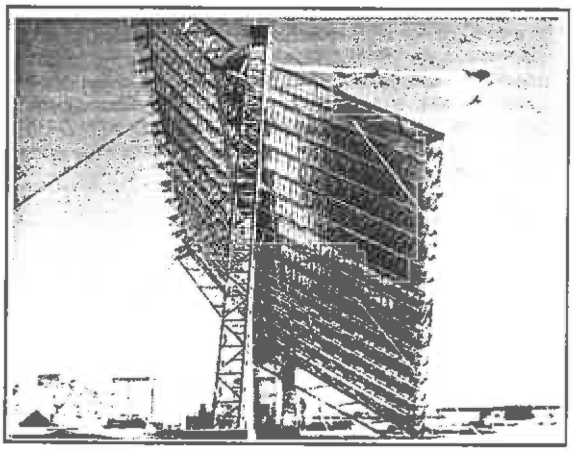

FIGURE 1. PKI 300 m$^2$ COLLECTOR AT SANDIA NATIONAL LABORATORY, ALBUQUERQUE, NEW MEXICO

COLLECTOR CHARACTERISTICS

The first PKI 300 m$^2$ Square Dish solar collector has been tested at the Distributed Receiver Test Facility at Sandia National Laboratories, Albuquerque (Sandia) during 1988, see Figure 1 [13].

Table II shows collector performance for five of the test days. During most of the test period, the collector was configured to deliver saturated steam (at 273°C) to an oil-fired superheater and then to a reciprocating steam engine [14,15]. Since the quality of the steam out of the receiver could not be ascertained in this arrangement, this value has been arrived at indirectly. In a separate test, the heat added by the superheater was measured and assumed constant. The heat delivered to the engine was integrated for the period and that added by the burner subtracted from the total. Derived performance of the collector and piping is in the 0.62 to 0.78 range, (heat delivered/incident solar on the collector area).

| Date | Period | Integrated Incident Solar kW | Insolation Average kW/m2 | Heat Delivered to Engine kW | Oil Burner Contribution kW | Attributed to Collector kW | All Day Effic'y |
|---|---|---|---|---|---|---|---|
| 9-May-88 | 6:20 to 17:47 | 3042 | .900 | 2985 | 736 | 2249 | .74 |
| | 12:20 to 14:20 | 567 | .961 | 553 | 128 | 425 | .75 |
| 20-Jun-88 | 6:18 to 17:30 | 2757 | .834 | 2671 | 697 | 1934 | .70 |
| 17-Jul-88 | 6:41 to 16:06 | 2413 | .844 | 2273 | 602 | 1671 | .69 |
| 22-Jul-88 | 6:44 to 15:52 | 1925 | .715 | 1965 | 578 | 1387 | .72 |
| 21-Aug-88 | 9:44 to 15:44 | 1503 | .857 | 955 | --- | 955* | .64 |

* Solar only operation

TABLE II: PKI COLLECTOR/PIPING PERFORMANCE TEST RESULTS OF UNIT INSTALLED AT SANDIA

Some of the collector piping was anchored directly to structural members (two places) which contributed to higher than desired heat losses from the piping. In addition, the piping, including the fittings, valves and pipe hangers, in the auxiliary boiler and from there to the engine was inadequately insulated. In a configuration for steam delivery into field piping, the measured performance would be enhanced by an estimated 6 kW. The ordinary glass mirrors used on this collector had a reflectivity of 0.86. With low iron glass mirrors with a reflectivity of 0.95 to 0.96 now available, the net collector efficiency at 300°C is estimated to be 0.87 at 0.868 kW/m$^2$ insolation as shown in Table III.

| | Loss Mechanism | 300°C | 350°C | 400°C | 450°C | 500°C | 550°C |
|---|---|---|---|---|---|---|---|
| Cavity | Qrad | 2.7 | 3.9 | 5.1 | 7.0 | 9.1 | 11.9 |
| | Qconv* | 3.8 | 4.4 | 5.1 | 5.8 | 6.6 | 7.4 |
| | Qcond | 0.4 | 0.5 | 0.6 | 0.7 | 0.7 | 0.9 |
| Aperture | QTotal* | 5.2 | 5.2 | 5.2 | 5.2 | 5.2 | 5.4 |
| Piping | Qcond | 2.3 | 2.6 | 3.0 | 3.4 | 3.7 | 4.2 |
| Losses | TOTAL kWth | 14.4 | 16.6 | 19.0 | 22.1 | 25.3 | 29.8 |
| Collector:** | @ 1kW/m2 | 258 | 256 | 254 | 250 | 248 | 243 |
| Delivery | @ 0.868 kW/m2 | 222 | 220 | 218 | 215 | 212 | 207 |
| Peak Efficiency | | 88% | 87% | 86% | 85% | 84% | 82% |
| Average Efficiency*** | | 87% | 86% | 85% | 84% | 83% | 81% |
| Annual Efficiency **** | | 85% | 84% | 83% | 82% | 81% | 79% |

* Wind: 10 m/s

** Concentrator Mirror reflectivity = 0.925 (0.95 -0.025 dirt allowance)

*** Operating @ 868 W/m2 Power Weighted Average solar energy in during operation (>450 W/m2), see text.

**** Approximately 2% of Direct Normal insolation is below the operating parameter of 450 W/m2 for Square Dishes.

These calculations were based on Ref. [16]

TABLE III: ESTIMATED COLLECTOR LOSSES AT VARIOUS TEMPERATURES: STEAM DELIVERY @ GROUND LEVEL: PKI 300 m$^2$

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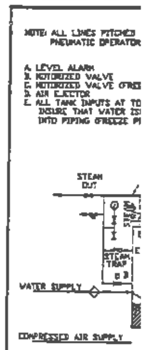

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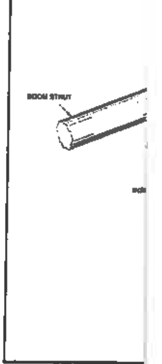

FIGURE 3. M

## WATER MANAGEMENT

When using water/steam as the heat transfer medium to transport thermal energy from a distributed field of point focusing solar collectors, water management is important. Good control of the water inventory minimizes parasitic power requirements, the thermal mass of the system, and thermal losses when the system is not operating. A three collector water management package (Figure 2) has been developed which controls the

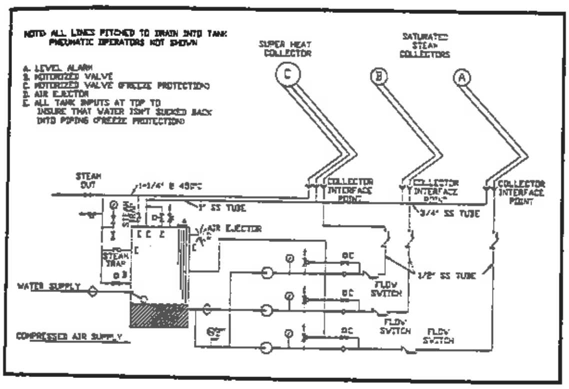


FIGURE 2. WATER MANAGEMENT SYSTEM

feedwater flow of two identical monotube receivers (Figure 3), so that dry saturated or slightly superheated steam is produced and delivered to a similar superheating receiver (on the third collector). This collector, which also has its own feedpump, delivers superheated steam to the manifold. The ratio of the number of saturated steam producing collectors to a superheating one is based on the ratio of enthalpy increase in bringing water from 80°C to dry saturated steam at the pressures of interest, (Table IV). Other factors involved in establishing this ratio are the desire to limit field losses by having the fewest collectors producing superheated steam (a ratio of 3:1 would do this), and the requirement for control and stability of the superheating receiver. Control is enhanced by the superheating collector producing part of its saturated steam. During startup, in fact, it operates as a saturated

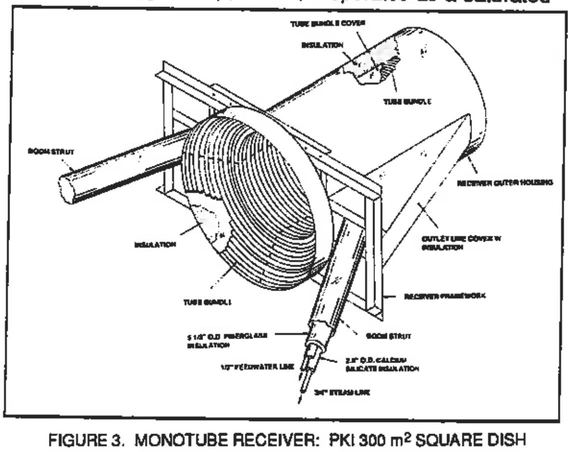

FIGURE 3. MONOTUBE RECEIVER: PKI 300 m$^2$ SQUARE DISH

steam collector until flows of steam have been established and the connecting piping heated. The front one third of the superheating receiver is identical to the others since the feedwater flow is used to cool the aperture coil which intercepts the spurious flux around the cavity and also protects the receiver housing during sun acquisition and walkoffs.

Operation of the speed control of the pumps and temperature setpoints on the water bypass valves are based on the system developed and tested for use with the 40 kWe steam engine [16]. Just as in operating a steam turbine or engine, it is important to eliminate any liquid water from being injected into the steam manifold. In this case the hot water is returned to the feedwater pump which recycles the heat back to the receivers. Water hammer in the larger steam lines is also eliminated as is the potential for the water freezing in the extended field piping. In a large operating system, the late startup of a collector set could have a severe impact on the quality of the superheat of the entire field if water were allowed to be injected into the manifold.

| Central Engine Nominal, $MW_e$ | °F | Psi | Nominal Efficiency* | Enthalpy 180°-Sat | Btu/lb SH | Ratio |
|---|---|---|---|---|---|---|
| 0.5 | 750 | 700 | 0.24 | 1051 | 321 | 3.3 : 1 |
| 10 | 900 | 900 | 0.30 | 1048 | 256 | 2.4 : 1 |
| 50 | 1000 | 1000 | 0.33 | 1043 | 312 | 3.3 : 1 |

* Taken from [17]

TABLE IV: STEAM TO ELECTRICITY CENTRAL PLANT TURBINE SIZE, PERFORMANCE, AND STEAM CONDITIONS

A single controller will operate the three collectors and monitor their operation. The controller includes a communication device for remote control and monitoring from a central facility. Where freeze protection is required, the water management system would be placed below grade and the receivers and piping constructed so that they drain into the single feedwater tank for each three collector set.

In a monotube boiling water receiver like the one tested with the collector at Sandia (Figure 3), the output temperature is controlled by varying the speed of the boiler feed pump. In this installation, the speed of the pump is actively adjusted by the collector controller through an algorithm which includes insolation (available from the sun-tracking sensors) and output temperature. Two programmable gains have been included to enable an operator to adjust response to the thermal characteristics of the system. Too high a gain results in temperature oscillations while too low a gain slows response of the system so that optimal temperatures for best engine performance lag behind changes in sun levels.

## FIELD PIPING THERMAL RESPONSE and LOSSES

Figure 4 illustrates that in a 10 MWe size system, with insolation at 450 W/m$^2$, field piping contributes only 8 minutes (of an estimated 18 minutes) of energy needed to preheat the piping to enable delivery of superheated steam to the central engine or process. For an average day in Albuquerque, the heat capacity of this thermal transport loop is only 0.8% of the collected energy. The heat lost through the same piping during a one hour cloudy period requires only 1-1/2 minutes of collector energy output to replenish the loss, although it would take additional time to reheat the receivers before beginning this process.

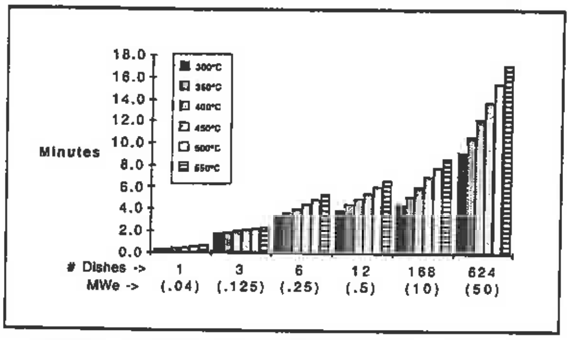


FIGURE 4. STARTING TIME EQUIVALENT @ 475 W/m$^2$ TO FIELD HEAT CAPACITY vs. FIELD SIZE FOR VARIOUS FIELD TEMPERATURES

Note in Figure 5 that the two 40 meter runs of pipe with doglegs to accommodate thermal expansion carry the steam from the two collectors which produce saturated steam to the third which does the superheating.

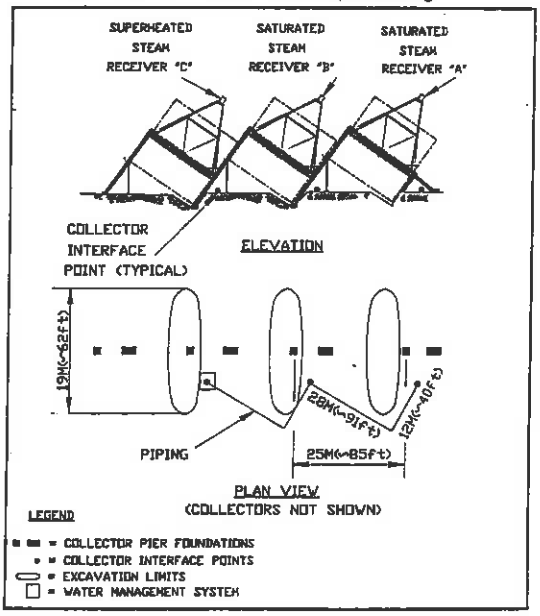


FIGURE 5. THREE COLLECTOR PIPING LAYOUT

Heat loss calculations use characteristics from insulation tables and the values are consistent with those included in [12]. Using lessons learned in the various DOE projects, two or more layers of insulation of equal thickness are utilized with joints staggered so no thermal short circuits develop with the motion caused by thermal expansion. For the saturated lines, both layers are made of fiberglass except for the pipe support areas. For superheated lines and regions of pipe supports, calcium silicate is used for the inside layer (outside layer also in the regions of the pipe supports in both cases).

It is noteworthy that even at 550°C, instantaneous field piping losses can be kept below 4% of the energy collected (Figure 6).

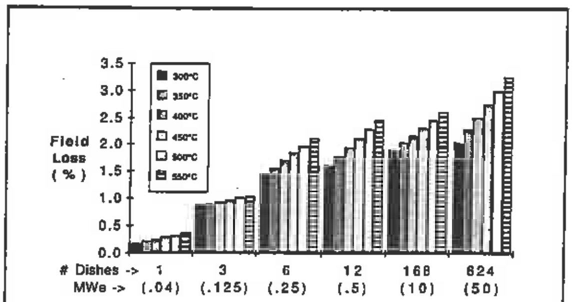


FIGURE 6. FIELD LOSSES vs. FIELD SIZE FOR VARIOUS OUTPUT TEMPERATURES

A requirement of operating a loop with low thermal mass is to minimize the inventory of water in the system. This is accomplished by keeping the steam lines empty of water at all times and by the proper programming of the startup procedure. Figure 4 shows the times required to bring the various sized fields to operating temperature with the insolation at 450 W/m$^2$. Less than 2% of the direct normal insolation occurs below this low insolation value as can be seen in Figure 7. This figure was generated by taking the typical meteorological year (TMY) data for Albuquerque, NM, and totaling the amount of direct normal sunlight which occurs in the region centered at the value listed.

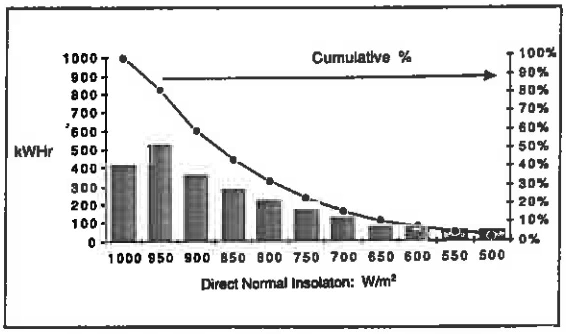


FIGURE 7. ALBUQUERQUE TMY ENERGY AVAILABLE TO SOLAR COLLECTORS



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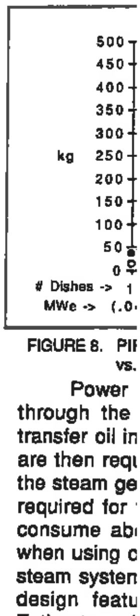


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One way to arrive at an annual field performance for a more realistic indication (instead of at peak insolation) of what part of the energy collected is lost in the field is to use the values from Figure 7 to evaluate the energy available. Since the rate of heat loss from the field piping at the operating temperatures is a constant no matter what the insolation, the percent lost in the piping goes up for lower values of insolation. Average insolation during operation occurs at 868 W/m$^2$ and Figure 6 shows the percent of the heat lost at this value for the various fields. This value is the power weighted annual field loss during operation of this heat transport approach.

Figure 8 shows that the weight of the pipes in the transport piping per collector increases with field size and in a 10 MWe field, 230 kg of steam delivery piping is required per collector. In a system which uses a heat transfer fluid, two sets of much larger pipes are required (though not at as high a pressure) for the same collector aperture than is required for steam. For the steam system, the supply piping, at low pressure, weighs less than half the steam piping. The piping thermal losses per collector for the various size fields are also given.

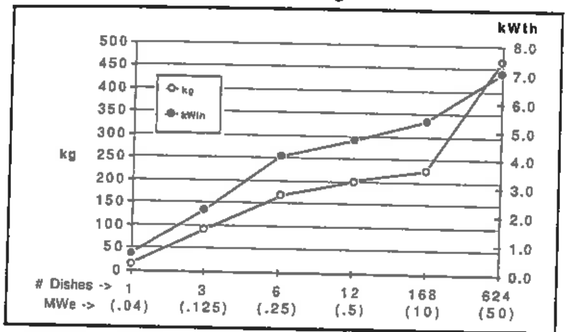


FIGURE 8. PIPING WEIGHT AND THERMAL LOSS PER COLLECTOR vs. FIELD SIZE

Power is needed to pump the heat transfer medium through the collectors and field piping. With a heat transfer oil in Shenandoah, 16 kWe are required. 9 kWe are then required to pump the fluid from storage through the steam generator, [4]. In addition, another 9 kWe are required for the boiler feedwater pump. Totaled, these consume about 23% of the generated electrical power when using only the solar collectors in this Syltherm fluid/steam system. Without a separate storage tank and other design features, this parameter could be reduced. Estimates of the Luz' SEGS III plant indicate that the transport fluid and feedwater pumps require around 6 to 10% of the electrical power output of the plant.

Three years of testing the variable speed motor/pump for the Molokai Small Community Solar Experiment module both during development and at Sandia have shown that the power required to operate the feedwater pump for one collector varies from 0.3 to 1.5 kWe depending on the system operating pressure and insolation level. This parasitic load amounts to 3.1% of a 40 kWe power plant (@18% thermal to electric), to less than 2% of the electrical output of the generated power in a 10 MWe plant (@ 30%). Because the unit pumps only the volume required to produce steam at the required temperature, this parasitic load varies with insolation which minimizes the power required throughout the day.

## CONCLUSION

In the design of a solar hardware system, the goal is to ***deliver the most usable energy per year for the least money***, i.e., collect as much energy as possible, deliver it without storage or waste, and minimize electrical parasitics. Higher quality energy leads to greater return on equipment investment.

Steam technology for the collection and transport of thermal energy is simple and very well known. Its direct use in large solar collectors and subsequent piping to the point of use is straightforward and has been shown to have very high performance as a transport medium.

In the 10 to 50 MWe systems, which involve from 168 to 624 collectors and their field piping:

- The Square Dish fields deliver over 80% of the energy available;
- Less than 4% of energy collected is lost in the transport piping;
- Only 0.8% of the average energy collected in a day is stored in the field piping;
- Less than 2% of the electrical power generated is required to pump water for the steam.

Figure 9 shows that at the various steam delivery temperatures, the collector field performance, including both the conversion of solar radiation to heat and its delivery to a central point, varies between 86% (@ 300°C) and 83% (@ 550°C). These values are for steady state operation and do not include the dynamic interactions between the site specific weather and the equipment or process which utilizes the energy. It should be noted that two of each three solar collectors operate at the saturated steam temperature and the third operates at the higher superheated steam temperature.

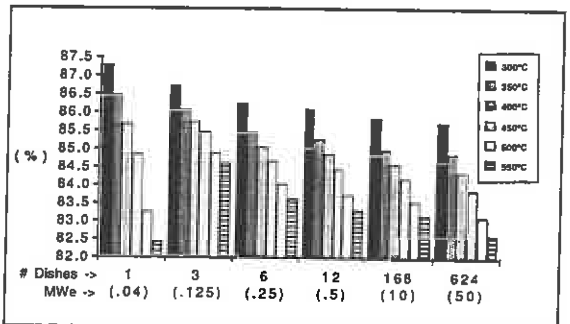


FIGURE 9. COLLECTOR WITH FIELD PERFORMANCE vs. FIELD SIZE FOR VARIOUS OUTPUT TEMPERATURES

Conscious of Jim Leonard's concept of the heat capacity tax, the system thermal response has been minimized. Because the steam pipes are small and start out operation in the morning with a vacuum inside, the time required for the energy to bring the field piping to operating temperature from a cold start is only minutes in a small field, and less than 20 minutes in a larger field using the marginal sun in the early morning. This makes the field very responsive to the sun and maximizes delivery of energy to the process at useful temperatures.

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