DOE/CS/32199--T1

DO NOT MICROFILM
COVER

FOSTER WHEELER DEVELOPMENT CORPORATION

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

DOE/CS/32199--T1

DOE/CS/32199--T1

DE82 008537

CONCEPTUAL DESIGN REPORT

FOR

SOLAR PRODUCTION OF INDUSTRIAL PROCESS STEAM
RANGING IN TEMPERATURE FROM 300°F TO 550°F
(PHASE 1)

AC03-7PCS 32199

Period of Performance

September 30, 1978 to June 30, 1979

DOE Contract No. ET-78-C-03-2199

FWDC No. 9-41-4010

By

Gopal D. Gupta, Project Manager

NOTICE

January 5, 1979

PORTIONS OF THIS REPORT ARE ILLEGIBLE. It
has been reproduced from the best available
copy to permit the broadest possible avail-
ability.

FOSTER WHEELER DEVELOPMENT CORPORATION

12 Peach Tree Hill Road, Livingston, New Jersey 07039

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

Handwritten signature/initials

The contents of this report have been reviewed and approved by:

R. J. Zoschak
Technical Director, Applied
Thermodynamics Research

A. C. Gangadharan
Manager, Engineering Science
and Technology Department

LEGAL NOTICE

This report was prepared by Foster Wheeler Development Corporation as an account of work performed for the Department of Energy. Neither Foster Wheeler Development Corporation nor any person acting on behalf of Foster Wheeler Development Corporation:

(a) makes any warranties or representation, express or implied, that the use of any information, apparatus, method or process disclosed in this report may not infringe privately owned rights; or

(b) assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

TABLE OF CONTENTS

<u>Section</u>	<u>Page</u>
1 INTRODUCTION AND SUMMARY	1-1
1.1 Background	1-1
1.2 System Cost-Effectiveness Criterion	1-3
1.3 Summary	1-4
2 COLLECTOR	2-1
2.1 Evaluation and Selection	2-1
2.2 Site Selection	2-6
3 SYSTEM OPTIMIZATION	3-1
3.1 Storage vs. No Storage	3-1
3.2 Optimization of System Alternate 1	3-2
3.2.1 Boiler	3-2
3.2.2 Preheater	3-5
3.2.3 Heat-Transport Fluid	3-7
3.2.4 Fluid Temperature	3-10
3.2.5 Pipe Size	3-14
3.2.6 Optimized System Alternate 1	3-17
3.3 Optimization of System Alternate 2	3-17
3.3.1 Flash Boiler	3-19
3.3.2 Freeze Protection	3-19
3.3.3 Circulation Ratio	3-21
3.3.4 Pipe Size	3-24
3.3.5 Optimized System Alternate 2	3-24
3.4 Optimum Design of System Alternate 3	3-26
3.4.1 Tube Dry-Out	3-28
3.4.2 Corrosion	3-29
3.4.3 Flow Instability	3-31
3.4.4 Circulation Ratio	3-32
3.4.5 Receiver Tube Configuration	3-32
3.4.6 Number of Collector Rows	3-35
3.4.7 Collector Inlet and Outlet Pipe Size	3-36
3.4.8 Optimized System Alternate 3	3-36
3.5 Comparison Between Alternate Systems	3-38

TABLE OF CONTENTS (Cont)

<u>Section</u>		<u>Page</u>
4	ENVIRONMENTAL IMPACT ASSESSMENT	4-1
	4.1 Historical	4-1
	4.2 Topography and Soils	4-1
	4.3 Hydrology and Water Quality	4-1
	4.4 Odor	4-2
	4.5 Air Quality	4-2
	4.6 Noise	4-2
	4.7 Visual	4-2
	4.8 Community Services and Facilities	4-3
	4.9 Wildlife	4-3
5	PRELIMINARY SAFETY ANALYSIS	5-1
	5.1 Introduction	5-1
	5.2 Alternative Conceptual Designs	5-1
	5.3 Potential Safety Problems	5-2
	5.3.1 Release of an Organic Heat Transfer Fluid to the Atmosphere	5-2
	5.3.2 Overpressurizing the System	5-4
	5.3.3 Overheating the Collectors	5-5
	5.3.4 Corrosion	5-5
	5.3.5 Boiler Hazards	5-6
	5.4 Conclusions of the Preliminary Safety Analysis	5-6
6	REFERENCES	6-1

LIST OF ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
2.1	Plan View of Dow's Dalton Plant	2-8
2.2	Topographic Map of Dow's Dalton Plant	2-9
3.1	Schematic--System Alternate 1	3-3
3.2	Annual Collected Energy Variation With Average Fluid Temperature Through the Collector	3-12
3.3	Schematic--System Alternate 2	3-18
3.4	Pressure Drop vs. Circulation Ratio (Flow Rate)	3-23
3.5	Schematic--System Alternate 3	3-27
3.6	Alternate Receiver Configurations	3-33
3.7	Side View of the Receiver Configurations	3-34

LIST OF TABLES

<u>Table</u>		<u>Page</u>
2.1	Concept Comparisons of Distributed Collector Systems	2-2
2.2	Cost-Effectiveness of Various Collector Systems	2-4
3.1	Comparison of Heat-Transport Fluids	3-8
3.2	Fluid Temperature Effect on Boiler Cost, Pump Cost, and Collector Efficiency	3-11
3.3	Cost-Effectiveness vs. Boiler Outlet Temperature (Boiler Inlet Temperature = 525°F)	3-15
3.4	Cost-Effectiveness vs. Pipe Size	3-16
3.5	Cost Effectiveness of Various Circulation Ratios	3-22
3.6	Cost-Effectiveness vs. Pipe Size	3-25
3.7	Flow Patterns Predicted From Baker Flow Map for Various Steam Quality and Mass Velocity Values	3-30
3.8	Effect of Circulation Ratio on Collector Outlet Steam Quality	3-32
3.9	Cost-Effectiveness of the Three Receiver Tube Configurations	3-35
3.10	Cost-Effectiveness vs. Inlet Pipe Size (Outlet Pipe Nominal Diameter = 4 in.)	3-37
3.11	Cost-Effectiveness vs. Outlet Pipe Size (Inlet Pipe Nominal Diameter = 3 in.)	3-37
3.12	Comparison of the Three Alternate System Concepts	3-39
5.1	Properties of Heat Transfer Fluids	5-3

SECTION 1
INTRODUCTION AND SUMMARY

Section 1

INTRODUCTION AND SUMMARY

1.1 BACKGROUND

This is Foster Wheeler Development Corporation's Conceptual Design Report for the Department of Energy project entitled, "Solar Production of Industrial Process Steam." The conceptual design is part of the Phase 1 studies in the overall project. The objectives of Phase 1 are:

- Design a cost-effective solar steam generating system, using state-of-the-art components and technology, to supply steam for Dow Chemical Company's Dalton, Georgia plant
- Predict the performance of the solar steam plant
- Conduct a safety evaluation and an environmental impact assessment of the solar system
- Conduct an economic analysis to determine the potential economic benefits of a solar-augmented process steam production system compared with an existing fossil-fuel-fired steam generator
- Promote the project extensively to make it visible to industry and the general public.

The conceptual design began with studies and evaluations of various alternative design configurations. From these evaluations a final candidate design configuration was selected which will be used for further detailed analysis and engineering.

An evaluation of various types of distributed collector systems is presented in Section 2. The tower-type central receiver and flat-plate concepts

were omitted from this evaluation. Flat-plate concepts are generally not efficient above 250°F and tower-type central receivers are not cost-effective below 700°F. Cost-effectiveness of various distributed collectors was determined on the basis of total relative cost (including foundation and installation) of the collector per square foot of collector surface for the same thermal output at noon. The considerations used to select the optimum site also are given in Section 2.

System optimization studies are presented in Section 3. Whether a storage facility should be provided is first discussed. Three alternate system configurations are then studied. These are:

- System Alternate 1. An organic oil is heated in the collectors and steam is generated in a boiler.
- System Alternate 2. Pressurized water is heated in the collectors and steam is generated in a flash tank.
- System Alternate 3. Water is partially boiled in the collectors and steam is separated in a steam drum.

These three systems are described and optimized in Subsections 3.2, 3.3, and 3.4, respectively. The three alternate system configurations are compared and a final concept is selected in Subsection 3.5. A cost-effectiveness criterion based on the unit cost of net annual thermal energy generated was used in the optimization studies. A discussion of this cost-effectiveness criterion is provided in Subsection 1.2.

The impact of the system on the environment is assessed in Section 4 and a preliminary safety analysis is presented in Section 5. Both studies were

utilized in the selection of the final concepts. A list of references is provided in Section 6.

1.2 SYSTEM COST-EFFECTIVENESS CRITERION

A system cost-effectiveness criterion was developed to provide a basis for optimizing the three alternate system configurations and selecting the final concepts. Unit cost of net annually generated thermal energy was used as this criterion.

In comparing the system alternates, the cost of only those components which are different in each system were estimated. The cost of the base configuration (common items) was then added to obtain the total cost of the system.

Considerable thought was given to how the net annually generated thermal energy should be determined. It is logical that annually consumed parasitic energy should be subtracted from annually collected solar energy. However, it is not clear whether electrical parasitic energy should be treated similarly or differently as thermal parasitic energy. Since electrical energy is produced from fossil fuels with an efficiency of only 30 to 35 percent, it does not seem reasonable to treat the two forms of energy in the same manner. In this report, electrical parasitic energy was converted into equivalent thermal energy by using a multiplication factor of 3. This is further justified by the fact that the cost of electrical energy at Dalton, Georgia varies between 2.6 to 4.6 times that of fossil energy depending on whether one uses oil or natural gas or a combination of the two. Therefore,

$$\text{Cost Effectiveness} = \frac{\text{Total Installed Cost}}{\text{Net Annually Generated Thermal Energy}}$$

Net Annually Generated Thermal Energy = Annual Solar Energy Collected - (Thermal Parasitic Energy + 3 x Electrical Parasitic Energy)

Thermal parasitic energy consists only of the thermal energy required for freeze protection. Thermal losses are not included and have been assumed to be equal for all systems considered. The fraction of electrical parasitic energy that is returned to the system as low-grade heat (e.g., as a result of pump inefficiency) is also omitted.

1.3 SUMMARY

The following is a summary of the solar process steam system selected for Dow Chemical's Dalton Plant.

- Collector. SLATSTM collector, manufactured by Suntec Systems, Inc.; 10,000 ft²; one-axis tracking; east-west orientation; five rows; 200-ft-long, double-tube receiver in each row; unidirectional flow through receiver.
- Storage. None required.
- System Description. Water partially boiled in the collectors. Steam drum utilized to separate steam from water, feedwater fed directly into the steam drum, overnight freeze protection provided.
- Steam Drum. No steam dryer required.
- Piping
 - Collector inlet: 3 in. Sch. 40 insulated
 - Collector outlet: 4 in. Sch. 40 insulated
 - No heat tracing.
- Pump
 - Centrifugal pump, single speed, 2 hp motor
 - Annual Pump Energy Consumed $\approx 12.2 \times 10^6$ Btu electric

● Operating Conditions

- Feedwater temperature = 205°F
- Circulation ratio = 25
- Fluid flow rate = 92 gal/min
- Average temperature in collector = 367°F
- Steam quality at collector outlet = 3.6 percent
- Mass velocity through receiver = 434,780 lb/h·ft²
- Steam outlet pressure = 155 lb/in²g

● Freeze Protection

- Hot water circulating system
- Existing feedwater pump utilized
- Flow rate = 1.5 gal/min
- Estimated annual energy consumed = 25.9 x 10⁶ Btu

● Energy Production

- Annual thermal energy produced = 2247 x 10⁶ Btu

● Steam Production

- Design value = 1600 lb/h
- Annual value = 2186 x 10³ lb/yr

SECTION 2
COLLECTOR

Section 2

COLLECTOR

2.1 EVALUATION AND SELECTION

Several types of focusing collectors were compared to establish a suitable concept for solar steam production of Dow Chemical's Dalton, Georgia, plant. Comparison criteria included:

- Design features
- Development status
- Design complexity
- Thermal performance
- Land area utilization
- Drive/tracking complexity
- Relative cost.

Lens type collector systems were excluded from these comparisons because of their limited concentration ratio and resultant low efficiency at elevated temperatures. Table 2.1 compares four distributed collector system concepts:

- Segmented reflector in planar array
- Parabolic trough with moving receiver
- Parabolic trough with stationary receiver
- Paraboloid; fully tracking.

Table 2.1 Concept Comparisons of Distributed Collector Systems

Concept	Concentration Ratio	Maximum Fluid Outlet Temperature (°F)	Concept Features	Development Status	Design Complexity
Segmented reflector in planar array. Fixed receiver. Sun-elevation.	35-40	675	<ul style="list-style-type: none"> • Stationary receiver • Stationary structure • Large, continuous collector rows • Moving segmented reflector • Low power drive system. 	In production on project basis. Demonstrated high temperature operation.	Simple and repetitive modules. Stationary receiver allows rigid piping and connections. Control-tracking drive operates large collector area. Greater number of parts in module.
Cylindrical parabola. Receiver mounted to reflector. Sun-elevation or sun-hour tracking.	40-70	675	<ul style="list-style-type: none"> • Moving receiver • Moving reflector • Long, continuous rows • High performance potential. 	In production on project basis. Demonstrated high-temperature operation.	Repetitive modular construction. Simple sun-tracking drive operates large collector area. Moving receiver requires rotating couplings and flexible connections.
Cylindrical parabola. Receiver stationary. Sun-elevation or sun-hour tracking.	30-50	675	<ul style="list-style-type: none"> • Stationary receiver • Moving reflector • Long, continuous rows • High-performance potential. 	In production on project basis. Demonstrated high-temperature operation.	Repetitive modular construction. Simple sun-tracking drive operates large collector area. Stationary receiver allows rigid piping. Tracking axis eccentric to reflector mass.
Point focus parabola. Moving receiver. Two-axis tracking.	300 or higher	700 or higher	<ul style="list-style-type: none"> • Moving receiver • Moving reflector • Moderate collector area per driven module • High concentration ratio, high temperature capability • Can operate above 1000°F with liquid metals • Highest performance potential. 	In development.	Complicated two-axis sun-tracking drive. Each module requires independent drive. Moving receiver requires 2-axis rotating coupling and flexible connections. Requires double curvature reflectors and exposed module-to-module piping. Low land area utilization. Large profile requires substantial structure and ground support.

Assuming that all the concepts use the same reflective surface and are used under ideal insolation conditions throughout the year, they would be rated in the following order according to the amount of solar energy collected annually:

- Paraboloid collector
- Parabolic trough
- Segmented reflector array.

In reality, these concepts utilize different reflective surfaces and involve different fabrication methods, thus leading to different costs. Therefore, a fair comparison of these collectors should be based on their cost-effectiveness measured in terms of the total installed cost per unit of annual collectible energy. Among the concepts listed in Table 2.1, the first three were explored in detail. The paraboloid concept was not pursued further because it is still in development.

The cost-effectiveness of five specific collector systems is presented in Table 2.2. SLATS, manufactured by Suntec Systems, Inc., is a line-focusing collector. The remaining four systems shown are parabolic trough collectors. They are identified by the names of their manufacturers. The three collectors (Hexcel, Solar Kinetics, and Acurex) are parabolic troughs with moving receivers. The Del-Jacobs collector represents a parabolic trough with a stationary receiver.

The cost-effectiveness measured in Table 2.2 is based on the total installed cost of the collector per unit area for the same thermal output at noon.

Table 2.2 Cost-Effectiveness of Various Collector Systems

Collector System	Relative Thermal Performance at Noon*			Costs (\$/ft ²) [§]	Relative Cost	Net Cost Effectiveness [¶]
	Claimed Efficiency (%)	Demonstrated Efficiency (%) [†]	Relative Collector Area Needed			
SLATS	61.5	54.5	1.00	28.00	1.00	1.00
HEXCEL	62	61.5	0.89	26.50	0.95	0.84
Solar Kinetics	63	53	1.03	29.00	1.04	1.07
Acurex	53	Not Available	1.03	30.50	1.09	1.12
Del-Jacobs	62	60	0.91	26.50	0.95	0.86

*Based on 392°F Fluid Outlet Temperature.

†Sandia Laboratory Reports, References 1 and 2.

§Costs include collector, freight, foundation, and installation.

¶Based on total relative cost of collector per ft² area for the same thermal output at noon.

Thermal output at noon was used for this comparison because the noon efficiencies for all the collector systems are well tested and documented. The installed cost of each collector was developed with the help of the collector manufacturers. Table 2.2 shows that all the collector systems are within a narrow band of ± 14 percent in terms of their cost-effectiveness. In addition maintainability, ease of repair, and other institutional factors must be considered. Therefore, final selection of the collector could not be based solely on cost effectiveness. The SLATS collector system was selected. The reasons for this decision include:

- Small Business. Suntec Systems, Inc., is a small business. DOE has a directive to encourage small business.
- Nonparabolic Trough Collector. Among the seven industrial process steam projects sponsored by DOE in the range of 300 to 550°F, SLATS is the only collector system which does not use the parabolic trough concept. The choice of SLATS is consistent with the DOE program objective of promoting various promising collector systems in order to create a viable and widespread industrial base.
- Cost-Effectiveness. SLATS falls in the middle of the narrow spectrum of collector cost-effectiveness as shown in Table 2.2. Two of the parabolic trough systems are more cost-effective, and two are less cost-effective than SLATS. All four of these parabolic trough systems are represented in the present DOE industrial process steam program.
- Fixed Receiver. SLATS does not require flexible and/or rotating couplings between receivers and the fluid loop. There is no theoretical limitation on the number of modules per row. A fixed receiver also reduces the operating cost of tracking.
- Microprocessor Controls. SLATS enables a convenient collector/thermal system interface. The microprocessor can also be utilized for data acquisition and real-time performance monitoring and evaluation. The collector system has flexibility to utilize closed loop or open loop control for solar tracking.

- Reliability and Maintainability. The SLATS collector contains segmented concentrators. If a slat is damaged, the total module reflector surface does not need to be replaced. The mirror segments are easily replaceable.

2.2 SITE SELECTION

The Dow Chemical latex plant in Dalton, Georgia, is located about 28 miles south of Chattanooga, Tennessee, and 94 miles north of Atlanta, Georgia, just off I-75. The site is easily accessible by road as well as by railway. The plant is located at 34° 43' latitude.

The plan view, Figure 2.1, shows the relative location of the process-plant units, boiler house, fuel-oil tanks, water tank, collector field warehouse, and administrative building. Dow Chemical owns several hundred acres of land to the north and northeast of the existing facilities.

The proposed solar plant will be designed and built with the collector field located north of the warehouse. This general location is ideal in many ways:

- Substantial land is available north of this location for future expansion.
- The existing boiler house, water tank, and warehouse are ideally located for use in the solar plant.
- The solar collectors are distinctly separated from the process-plant units, allowing easy access to the solar plant for visits by the project team members, DOE, and the general public.

The particular location selection has been directed toward:

- Minimizing the piping costs

- Minimizing the site preparation
- Avoiding any shadow from the neighboring structures.

A land contour map, shown in Figure 2.2, has been utilized to accomplish this task. To avoid the early morning shadow of the water tank, the collector field has been located about 150 ft away from the tank.

Figure 2.1 Plan View of Dow's Dalton Plant

Figure 2.2 Topographic Map of
Dow's Dalton Plant

SECTION 3
SYSTEM OPTIMIZATION

Section 3

SYSTEM OPTIMIZATION

3.1 STORAGE VS. NO STORAGE

Thermal storage systems are generally expensive. Thus, a more economically viable solar steam system is one which does not contain or need thermal storage facilities. The need for thermal storage in a hybrid solar process steam system depends on:

- The steam capacity of the solar system
- The steam demand of the process
- The effects of solar system transients on the total system and the process.

The solar system to be installed at Dow Dalton will contain 10,000 ft² of collector area. At peak insolation conditions, it will generate 1,600 lb/h steam.

The solar system is designed to supply steam for the continuous stripping operation in Plant 2 of the Dow-Dalton plant. The steam required by this operation is approximately 4,000 lb/h. Therefore, if the solar steam supply is suddenly cut off due to a cloud cover, the fossil boiler would have to take the full load. The existing boiler at the plant can pick up the full load in about 10 min. During these 10 min, the total steam pressure would be somewhat lower than the required 150 lb/in²g which would have no significant effect on the stripping operation.

Based on these facts, the Dow-Dalton Solar Process Steam System will not contain any thermal storage.

3.2 OPTIMIZATION OF SYSTEM ALTERNATE 1

A schematic of this system is shown in Figure 3.1. It contains a heat transport loop which delivers the hot organic fluid from the collectors to the boiler and recirculates the fluid to the collectors via a circulating pump. Steam-generating equipment is used in which the hot fluid from the collectors furnishes the heat required to convert the feedwater from the existing plant steam system into saturated steam at 155 lb/in²g. The organic fluid loop also requires a dump tank, an expansion tank, and an inert gas blanketing system.

This system was optimized to select the most cost-effective operating conditions and components. These optimization studies are discussed below.

3.2.1 Boiler

The boiler produces saturated steam by exchanging heat between the hot heat-transport fluid and the feedwater from the plant steam system. Desirable features for the boiler are:

- Simplicity of design and fabrication
- Ease of control
- Reliability
- Low cost.

Once-through, forced recirculation, and natural circulation boilers were considered.

Figure 3.1 Schematic--System Alternate 1

The once-through boiler has a fast load response and, at first glance, appears simple and low in cost. However, it requires very pure water, because any impurities are deposited in the tubes. The once-through boiler also requires a high water/steam mass velocity to ensure proper flow distribution and stability. This requirement results in a high pressure drop and hence high feedwater pump power. This boiler type would also need to be custom designed.

The forced-recirculation boiler requires a pump, in contrast to the natural-circulation boiler, and hence has inherently lower reliability, higher power consumption, and additional maintenance. Forced recirculation is unnecessary at the low heat-flux densities and relatively low operating pressure anticipated in the proposed solar energy plant.

Natural-circulation boilers are either of the drum or kettle type. The drum type requires a tubular heating section separate from the steam/water-separating drum. The drum and heating sections must be connected by exterior downcomer piping, resulting in a relatively costly assembly. In the kettle type, tubes containing the hot heat-transport fluid are submerged in the boiler water that partially fills a horizontal, cylindrical pressure vessel. Steam/water separation takes place in the upper portion of the vessel above the water level. This boiler type is inexpensive and readily available from many manufacturers. It is particularly suited for the low operating pressure contemplated and has adequate response. Because of these desirable features the kettle-type natural-circulation boiler has been selected for the proposed solar steam plant.

3.2.2 Preheater

The steam at 155 lb/in²g (368°F) from the feedwater available at 205°F can be produced either by heating the feedwater in a preheater to a maximum of about 365°F ensuring that no boiling occurs, and then generating steam in a boiler; or by injecting the feedwater directly into the boiler, eliminating the preheater completely.

The first configuration yields a higher Log Mean Temperature Difference (LMTD) across both the preheater and the boiler and reduces the net heat-transfer surface area. The second configuration develops into a simpler system, in which the installation, piping, and connection costs would be considerably less. However, the boiler surface area in the second configuration is larger. The cost of a larger, single boiler can still turn out to be lower than a two-unit system because the cost per unit surface area of the heat exchanger is reduced as the unit size increases. The following example illustrates this point. For purposes of this calculation the installation and piping costs are assumed to be the same in both cases. The fluid temperatures selected in this example are such that the average temperature of fluid through the collectors remains unchanged.

Case 1 - Boiler With Preheater

Assumed:

Temperature of fluid at boiler inlet = $T_1 = 500^\circ\text{F}$

Temperature of fluid at preheater outlet = $T_2 = 400^\circ\text{F}$

Duty of boiler	= 1.37×10^6 Btu/h
Duty of preheater	= 2.77×10^5 Btu/h

Average heat-transfer coefficient
in boiler = 153 Btu/h·ft²·°F*

Average heat-transfer coefficient
in preheater = 94.9 Btu/h·ft²·°F*

Calculated:

Temperature of fluid at boiler outlet = 417°F

Boiler LMTD = 83.8°F

Boiler surface area = 107 ft²

Preheater LMTD = 105.7°F

Preheater surface area = 28 ft²

Boiler cost = \$2,525[†]

Preheater cost = \$1,120[†]

Total boiler and preheater cost = \$3,645

Case 2 - Boiler Only (no preheater)Assumed:

Temperature of fluid at boiler inlet = 500°F

Temperature of fluid at boiler outlet = 400°F

Duty of boiler = 1.647 x 10⁶ Btu/h

Average heat-transfer coefficient in
sensible heat section = 94.9 Btu/h·ft²·°F*

Average heat-transfer coefficient in
latent heat section = 153 Btu/h·ft²·°F*

Calculated:

Boiler surface area = 146 ft²

Boiler cost = \$3,270[†]

The above computations show that Case 2 (boiler only) is slightly more cost-effective than Case 1 (boiler with preheater), even if the piping and installation costs are assumed to be the same in both cases. Based on this economic evaluation, the System Alternate 1 design contains only one heat exchanger, namely, the boiler.

*Based on vendor data.

†Based on vendor cost data.

3.2.3 Heat-Transport Fluid

Cost factors that must be considered in selecting the heat-transport fluid are:

- Initial charge of fluid
- Replacing or reconditioning thermally decomposed fluid
- Heating and/or recirculating fluid on cold nights to keep it from becoming too viscous in the morning, i.e., to keep it "pumpable"
- Pumping (pump power requirements).

A comparison of the relevant properties and costs of a number of promising heat-transport fluids is presented in Table 3.1. The maximum thermal stability temperature values in this table correspond to an acceptable upper limit of 10 percent decomposition level of the fluid in 214 exposure weeks. The total number of exposure weeks are computed on the basis of 0.5 exposure hour per operating hour, 8 operating hours per day and 300 days per year of operation.

From the thermal stability point of view, all of the heat-transport fluids would be adequate if the maximum temperature of the fluid is maintained below 500°F. However, if a maximum fluid temperature of 525°F appears to be more cost-effective from other system considerations, only Caloria HT-43 and Therminol-55 would be unsuitable. If Caloria HT-43 or Therminol-55 were used at 525°F, they would need to be replaced approximately every 10 years.

Caloria HT-43 is the least expensive of the heat transport fluids listed in Table 3.1. However, it becomes highly viscous at ambient temperatures, reaching a viscosity of 300 cP at about 30°F. Piping and receiver temperatures

Table 3.1 Comparison of Heat-Transport Fluids

Fluid	Manufacturer	Maximum Thermal Stability Temperature* (°F)	Minimum Pumping Temperature (°F)	Specific Heat† (Btu/lb°F)	Viscosity† (ft ² /h)	Density† (lb/ft ³)	Cost (\$/gal)
Dowtherm G	Dow Chemical	640	20	0.478	0.0261	60.00	8.50
Dowtherm LF	Dow Chemical	530	-25	0.520	0.0166	55.54	8.86
Dowtherm J	Dow Chemical	540	-100	0.595	0.0450	44.36	6.00
Caloria HT-43	EXXON	500	30	0.600	0.0577	45.60	1.25
Therminol 55	Monsanto	500	25	0.620	0.0455	47.87	2.75
Therminol 60	Monsanto	525	-35	0.543	0.0238	53.30	6.90
Therminol 65	Monsanto	560	40	0.530	0.0350	53.70	8.48

*For a 30-year lifetime.

†At 400°F.

can drop below 30°F on cold nights, making it extremely difficult to get the Caloria circulating properly in the morning. Therefore, it would be necessary to use heat tracing or to heat and recirculate the fluid during the night. It was estimated that an additional capital cost of \$2,200 would be incurred for heat tracing. As indicated in Table 3.1, the use of Dowtherm LF, Dowtherm J, or Therminol-60 would eliminate the need for heat tracing. The extra initial cost of these three fluids would be \$2,660, \$1,660, and \$1,980, respectively as compared to Caloria HT-43. Therefore, the use of Dowtherm J or Therminol-60 would be more cost-effective than Caloria based solely on the initial capital cost. If the energy consumed in heat-tracing operation is taken into account, even Dowtherm LF will be more cost-effective than Caloria HT-43.

The pumping power depends on the viscosity of the fluid. Dowtherm LF has the lowest viscosity at 400°F and thus would require the least pumping power and have the lowest operating cost. Therminol-60 is the next best candidate fluid from the pumping power standpoint.

Considering all factors mentioned above, the best heat-transport fluid is Therminol-60. The following is a summary of all advantages of Therminol-60.

- Maximum Temperature. Can be operated at temperatures up to 525°F without needing replacement for 30 years.
- Minimum Temperature. Minimum pumping temperature is -25°F. Would need no heat tracing or hot fluid recirculation during the night.
- Cost-Effectiveness. Cheaper than Dowtherm LF by 22 percent. More cost-effective than Caloria HT-43.
- Pumping Power. Requires less pumping power than all but Dowtherm LF. The difference in pumping power is estimated to be less than 5 percent.

- Low Vapor Pressure. Vapor pressure up to 600°F is below 15 lb/in²a. In comparison, vapor pressure of Dowtherm J at 525°F is about 94 lb/in²a, requiring a pressurized inert gas blanket.
- Toxicity. Therminol-60 is not a primary skin-irritant.
- Autoignition. Autoignition temperature of 835°F is substantially higher than the maximum system operating temperature of 525°F.

3.2.4 Fluid Temperature

Selection of the optimum fluid temperature necessitates consideration of boiler cost, collector efficiency, and pump capital and operating costs. A summary of this study is presented in Table 3.2. Four values of the fluid temperature entering the boiler in the range of 450 to 525°F in conjunction with a number of fluid temperature values leaving the boiler in the range of 380 to 460°F were considered. The cost differences are computed with respect to a base configuration in which boiler inlet fluid temperature is 500°F and boiler outlet fluid temperature is 400°F. The piping costs are assumed to be constant in this study.

The required boiler surface area changes with changes in fluid temperature conditions. The resulting change in boiler cost is shown in Table 3.2. An increase in either inlet or outlet fluid temperatures tends to reduce boiler cost. Thus, the inlet/outlet temperature combination of 450°F/380°F yields the most expensive boiler, whereas 525°F/460°F yields the least expensive.

Collector efficiency depends on the average temperature of fluid in the collectors. A variation of annual collected energy with respect to the average fluid temperature is shown in Figure 3.2. Total collector area assumed in this

Table 3.2 Fluid Temperature Effect on Boiler Cost, Pump Cost, and Collector Efficiency

Boiler Inlet Temperature (°F)	Boiler Outlet Temperature (°F)	Ccost Difference From Base (\$)			Annual Energy Collected (10 ⁶ Btu)	Relative Pump Power	Unit Cost of Net Annual Thermal Energy Collected (\$/10 ⁶ Btu)
		Boiler	Pump	Net (Pump + Boiler)			
450	380	+2050	+650	+2700	2162	3.04	165.6
	400	+800	+1550	+2350	2140	8.27	245.6
475	380	+1410	+50	+1460	2135	1.20	150.5
	400	+300	+600	+900	2115	2.40	162.6
	420	-140	+1300	+1160	2093	6.02	210.5
500	380	+970	-450	+520	2110	0.58	146.9
	400	Base	Base	Base	2087	1.00	151.7
	420	-460	+450	-10	2064	1.93	162.0
	440	-770	+950	+180	2038	4.52	194.8
525	380	+590	-750	-160	2082	0.33	146.5
	400	-260	-500	-760	2058	0.51	149.5
	420	-670	-250	-920	2032	0.85	154.2
	440	-980	+300	-680	2008	1.58	163.2
	460	-1190	+750	-440	1984	3.49	187.2

Figure 3.2 Annual Collected Energy Variation With Average Fluid Temperature Through the Collector

figure is 10,000 ft². Calculations are made on the basis of Dalton, Georgia, latitude of 34.9 deg. Since no insolation data for Dalton, Georgia, are available, Figure 3.2 is based on ASHRAE insolation data for an average day in Atlanta, Georgia. The amount of annual collected energy for various inlet/outlet combinations is included in Table 3.2.

As the boiler inlet/outlet temperature set is changed, the required fluid flow rate varies. This affects the capital and operating costs of the circulating pump. The relative pump power required in different cases is presented in Table 3.2. The differences in the pump costs shown in Table 3.2 assume centrifugal pumps in all cases and include motor and foundation costs.³

The final selection of the fluid temperature was based on the total capital cost per net production of a unit of thermal energy. For purposes of this study, the capital cost for the base configuration was assumed to be \$300,000. Net energy from the solar system is equal to the energy delivered to the fluid minus the parasitic power translated into equivalent thermal energy.

Thus, Net Annual Energy = Annual energy produced - 3 x annual pump energy

$$\text{Annual Pump Energy} = \text{Pump hp} \times 8 \text{ h/day} \times 300 \text{ days/year} \times \frac{2545 \text{ Btu}}{\text{hp-h}}$$

The last column in Table 3.2 shows the relative merit of the inlet/outlet temperature combinations. The cost per unit of net thermal energy decreases as the boiler inlet temperature increases and the boiler outlet temperature decreases. As stated previously, the maximum fluid temperature is limited by

thermal stability considerations. The boiler outlet temperature can theoretically be as low as the steam temperature (368°F). However, at that value, the corresponding boiler size and cost would become infinite. Table 3.3 shows the cost per net thermal energy with respect to boiler outlet temperature for an inlet temperature of 525°F. The optimum value of the boiler outlet temperature is 370°F.

3.2.5 Pipe Size

Selection of satisfactory pipe size requires consideration of piping cost and pump capital and operating costs. The nominal diameter of the pipe considered ranged from 1 to 3 in. A summary of this study is presented in Table 3.4. The boiler inlet/outlet temperatures of 525/370°F were used for this study.

The fluid velocity variation with respect to pipe size was first determined. Fluid velocity adversely affects the piping support structure and the pressure drop in the pipe. Standard industrial practice is to keep fluid velocity below 10 ft/sec.

Cost-effectiveness of various pipe sizes is determined in terms of net thermal energy produced annually. The pump electrical energy is translated into the equivalent thermal energy by multiplying by a factor of 3. Based on the results presented in Table 3.4, the nominal pipe diameter of 2 in. was selected for this system.

Table 3.3 Cost-Effectiveness vs. Boiler Outlet Temperature
(Boiler Inlet Temperature = 525°F)

Boiler Outlet Temperature (°F)	Cost Difference From Base (\$)			Annual Energy Collected (10 ⁶ Btu)	Relative Pump Power	Unit Cost of Net Annual Thermal Energy Collected (\$/10 ⁶ Btu)
	Boiler	Pump	Net (Pump + Boiler)			
400	-260	-500	-260	2058	0.51	149.5
380	+590	-750	-160	2082	0.33	146.5
375	+885	-775	+110	2087	0.30	146.1
370	+1970	-800	+1170	2093	0.27	145.9
369	+2590	-800	+1790	2094	0.27	146.2

FOSTER WHEELER DEVELOPMENT CORPORATION

REF.: ET-78-C-03-2199
DATE: January 5, 1979

Table 3.4 Cost-Effectiveness vs. Pipe Size

Nominal Pipe Diameter (in.)	Fluid Velocity (ft/s)	Cost Difference From Base (\$)			Relative Pump Power	Unit Cost of Net Thermal Energy Produced (\$/10 ⁶ Btu)
		Installed Insulated Piping	Pump	Net (Pump + Piping)		
1	16.7	-1650	+1550	-250	11.73	170.21
1-1/2	7.1	-800	+450	-450	1.92	146.86
2	4.3	Base	Base	Base	1	145.35
2-1/2	3.0	+1300	-125	+1075	0.75	145.47
3	2.0	+2400	-200	+2050	0.64	145.77

3.2.6 Optimized System Alternate 1

Boiler	- Kettle type No preheater Boiler surface area = 270 ft ²
Heat Transport Fluid	- Therminol-60
Piping	- 2 in. Sch. 40 insulated No heat tracing
Operating Temperatures	- Boiler inlet temperature = 525°F Boiler outlet temperature = 370°F Fluid flow rate = 45 gal/min
Pump	- Centrifugal pump motor = 1.75 hp Annual pump energy consumed $\approx 10.7 \times 10^6$ Btu electric
Energy Production	- Annual thermal energy produced = 2093 $\times 10^6$ Btu

3.3 OPTIMIZATION OF SYSTEM ALTERNATE 2

A schematic of this system is shown in Figure 3.3. In this system, pressurized water is circulated through the collectors and a flash boiler is utilized to generate steam. The feedwater is fed in the flash boiler with a level controller on the flash boiler which regulates the feedwater flow rate. The water pressure for this system is not so high as to affect the receiver geometry or its performance. The advantages of this system are:

- Simplicity
- The organic fluid, dump tank, expansion tank, and the inert gas blanketing equipment are eliminated
- The average temperature of the fluid in the collector is lower, thus increasing the collector efficiency significantly
- Low pressure drops through the piping, thus reducing piping costs.

Figure 3.3 Schematic--System Alternate 2

The disadvantages of the system are:

- Freeze protection is required
- The location at which flashing occurs is subjected to erosion, requiring special design consideration
- A bigger pump is required, thus increasing the parasitic power requirement.

This system was optimized to select the most cost-effective system operating conditions and components. These optimization studies are discussed below.

3.3.1 Flash Boiler

In this concept, the pressure of the hot water from the collectors is suddenly reduced to 155 lb/in²g resulting in a mixture of water and steam. Steam is then separated from the water. This can be accomplished either by flashing the pressurized water inside the flash boiler and then separating steam from water in the upper part of the boiler, or by flashing the pressurized water across a throttling valve and using the flash boiler only as a steam separator.

The second configuration, namely, throttling valve with a steam separator was selected because it is simple in design, and costs approximately the same as a single-unit flash boiler.

3.3.2 Freeze Protection

This system requires overnight protection against freezing. This can be accomplished by overnight draining, electric heat tracing on piping and

valves, and preheating the receiver before start-up, or by heating and recirculating the water during the night.

The major drawbacks of the first method are that it consumes high-grade electrical energy and requires an initial capital investment.

The second method consumes mostly thermal energy and a small amount of electrical energy for the circulating pump. A small capital investment is needed for control equipment. The existing feedwater pump will be utilized for recirculating water for freeze protection. The feedwater, after circulating through the collector field would be fed back to the suction side of the existing feedwater pump.

The two methods are compared below. Both are based on 450 h/yr with the freeze protection system in operation.

Method 1 - Electric Heat Tracing

Estimated total electrical energy consumed per year	= 1350 kWh
Equivalent thermal energy consumed*	= 13.8×10^6 Btu
Estimated capital cost of heat tracing	= \$2200
Cost per unit of net thermal energy produced	= $\frac{302,200}{2216-13.82} = \frac{\$137.23}{10^6 \text{ Btu}}$

Method 2 - Circulating Hot Feedwater

Flow rate	= 1.5 gal/min
Thermal energy consumed	= 25.9×10^6 Btu
Pump power consumed	= 0.2 kWh
Total thermal energy consumed	= 25.9×10^6 Btu
Estimated capital cost investment	= \$300
Cost per unit of net thermal energy produced	= $\frac{300,000}{2216-25.9} = \frac{\$137.1}{10^6 \text{ Btu}}$

*Equivalent Thermal Energy = 3 x Electrical Energy.

Based on this comparison, the second method was selected because it is slightly more cost-effective and requires less initial capital investment.

3.3.3 Circulation Ratio

This is the most important parameter in the design of System Alternate 2. Qualitatively, as the circulation ratio increases, the water flow rate through the piping and the receiver increases. This would increase the fluid pressure drop through the piping. However, the required drop across the throttling valve would decrease. An increase in flow rate would reduce the average water temperature through the collector, thus increasing the annual energy collected.

Five circulation ratios ranging from 10 to 30 were evaluated. This evaluation is summarized in Table 3.5. The flow rates are directly proportional to the circulation ratio. The average temperatures of the fluid in the collector vary from 400 to 379°F for circulation ratios of 10 to 30 respectively. Corresponding amounts of annual energy collected are given in Table 3.5. The variation of pressure drops through the piping, including the receiver and that required across the throttling valve with respect to the flow rate (and circulation ratio) is plotted in Figure 3.4. The total required pump head is also shown.

The relative pump power and the differences in capital pump cost³ are shown in Table 3.5. The net thermal energy generated annually is computed by subtracting the equivalent thermal energy consumed by the circulating pump, from the energy delivered by the solar collectors. The cost-effectiveness of each configuration is measured in terms of the unit cost of net annual thermal

Table 3.5 Cost Effectiveness of Various Circulation Ratios

Circulation Ratio	Flow Rate (gal/min)	Collector Performance		Circulating Pump		Cost per Net Thermal Energy Generated Annually (\$/10 ⁶ Btu)
		Average Temperature (°F)	Annual Collected Energy (10 ⁵ Btu)	Relative Pump Power	Pump Cost Difference (\$)	
10	37	400	2189	0.80	-150	161.8
15	55	390	2206	0.78	-200	159.6
20	73	384	2216	0.85	-100	161.3
25	92	381	2223	1.0	Base	166.3
30	110	379	2226	1.25	+200	176.4

Figure 3.4 Pressure Drop vs. Circulation Ratio (Flow Rate)

energy. The base cost is assumed to be \$300,000. Based on the results presented in Table 3.5, the circulation rate of 15 was selected for this system.

3.3.4 Pipe Size

The selection of optimum pipe size requires a compromise between piping and pump costs. The optimization procedure followed is similar to the one described in Section 3.2.5 of this report. The range of pipe size considered in this study varied from 1 to 4 in. nominal diameter. A circulation ratio of 15 is used. A summary of this study is presented in Table 3.6. Based on these results, the optimum choice for nominal pipe diameter is 3 in.

3.3.5 Optimized System Alternate 2

- Flash Boiler Components
 - Throttling valve
 - Steam separator
- Piping - 3 in. Sch. 40 insulated; no heat-tracing
- Pump
 - Turbine pump, single-speed motor
 - Motor size = 17 hp
 - Annual pump energy consumed $\approx 104 \times 10^6$ Btu electric
- Operating Conditions
 - Circulation ratio = 15
 - Fluid flow rate = 55 gal/min
 - Collector inlet temperature = 358°F
 - Collector outlet temperature = 422°F
 - Throttling valve inlet pressure = 305 lb/in²g
 - Throttling valve outlet pressure = 155 lb/in²g

Table 3.6 Cost-Effectiveness vs. Pipe Size

Nominal Pipe Diameter (in.)	Fluid Velocity (ft/s)	Cost Difference From Base (\$)			Relative Pump Power	Unit Cost of Net Thermal Energy Produced (\$/10 ⁶ Btu)
		Installed Insulated Piping	Pump	Net		
1	20.42	-1600	+835	- 765	2.4	210.55
1-1/2	3.66	- 800	+ 85	- 715	1.17	164.13
2	5.26	Base	Base	Base	1.0	159.66
2-1/2	3.68	+1300	- 85	+1215	0.95	158.92
3	2.41	+2400	-465	+2235	0.92	158.64
3-1/2	1.78	+4300	-240	+4060	0.91	159.32
4	1.38	+6050	-290	+5760	0.907	160.13

- Freeze Protection
 - Hot water circulation system
 - Existing feedwater pump utilized
 - Flow rate = 1.5 gal/min
 - Estimated annual energy consumed = 25.9×10^6 Btu
- Energy Production
 - Annual thermal energy produced = 2206×10^6 Btu

3.4 OPTIMUM DESIGN OF SYSTEM ALTERNATE 3

A schematic of this system is shown in Figure 3.5. In this system, water is partially boiled in the collector and circulated through a steam drum where steam is separated from water. Feedwater is added to the steam drum at a rate which is controlled by the level controller on the steam drum. The same freeze protection system as used in System Alternate 2 has been selected for this concept. The advantages of this system are:

- The organic fluid, dump tank, expansion tank, and inert gas blanketing equipment are eliminated.
- The average temperature of the fluid in the collector is the lowest of the three alternate systems. Consequently, the highest collector efficiency would be expected.
- The pump size may be the same as in System Alternate 1. The parasitic power requirement is lower than for System Alternate 2.

The disadvantages of this system are:

- Freeze protection is required
- Dry-out (tube burn-out) is possible
- Flow instability may be encountered
- Corrosion of the receiver tubes may occur if the flow regime causes alternate drying and wetting.

Figure 3.5 Schematic--System Alternate 3

Optimization of this design involved selection of proper inlet and outlet pipe sizes, circulation ratio, number of parallel collector rows and the receiver configuration which would lower the required pump size without causing tube dry-out, corrosion, or flow instability problems. The three problem areas are discussed in the following subsections.

3.4.1 Tube Dry-Out

Tube dry-out is one of the most important limits on the design of a steam generator. This condition is characterized by a sharp reduction in the ability to transfer heat from a heated surface. This occurs when the liquid phase adjacent to the heat-transfer surface is replaced by vapor phase. This results in much higher tube metal surface temperatures. Thus the major advantage of having the lowest collector temperatures in System Alternate 3 would be lost if dry-out conditions prevailed.

A literature survey of analytical or experimental studies done in this area indicated that the majority of the existing information concerns vertical tube configurations, pressures in excess of 600 lb/in², and very high heat fluxes (over 100,000 Btu/h·ft²). In the present application, the tubes are horizontal, the pressure is low (170 lb/in²), and the heat flux is less than 10,000 Btu/h·ft².

The study most relevant to the present design conditions was conducted by Purnell and Novak.⁴ Horizontal tube tests are reported at pressures of 600 to 1200 lb/in². These tubes were heated from the top. A general conclusion of

the study was that the dry-out heat fluxes for horizontal tubes were one-third of those for vertical tubes. However, Purnell in a private communication indicated that this difference in horizontal and vertical tubes was observed because of the top surface heating of the horizontal tube. If the tubes are heated from the bottom, as is the case in the present application, the dry-out fluxes for the horizontal tubes are at least as high as those in the vertical tubes.

Another observation from the Purnell-Novak study is that for a constant mass fluid velocity, the dry-out fluxes decreased as the pressure was increased from 600 to 1200 lb/in². The same effect was observed for dry-out steam quality. This trend indicates a favorable effect of the lower pressure situation in the present application. All dry-out fluxes in Reference 4 were above 91,600 Btu/h·ft² and dry-out steam qualities were above 9 percent for the mass velocities in excess of 373,000 lb/h·ft². The present application involves maximum heat fluxes of about 10,000 Btu/h·ft² which is significantly lower than the critical flux.

To provide margins of safety for the present design, an upper value of 5 percent is set for the steam quality and a minimum value of 400,000 lb/h·ft² is set for the mass velocity. Eliminating any heat flux on the top of the tube would definitely diminish the possibility of dry-out.

3.4.2 Corrosion

Corrosion problems can occur only if there is alternate drying and wetting of the tube surface. During drying, any salts contained in water would concentrate at the tube surface and would lead to tube corrosion. Elimination

of heat flux on the top of the tube could diminish the likelihood that this problem would occur. Under two-phase flow conditions, the flow pattern in the tube primarily depends on the steam quality and the mass velocity. The Baker Flow Map⁵ was utilized in order to determine the flow pattern. The flow pattern predicted for mass velocities of 400,000 and 500,000 lb/h·ft² and steam qualities varying from 0 to 17 percent is presented in Table 3.7. The results indicate that the flow pattern below 5 percent steam quality is not expected to cause alternate wetting and drying. Therefore, corrosion of the receiver tubes is not anticipated.

Table 3.7 Flow Patterns Predicted From Baker Flow Map
for Various Steam Quality and Mass Velocity Values

Steam Quality (%)	Predicted Flow Patterns	
	Mass Velocity (400,000 lb/h·ft ²)	Mass Velocity (500,000 lb/h·ft ²)
0.1	plug	plug
0.5	slug	slug
1.0	slug	slug
2.0	slug	slug
3.0	slug	annular
4.0	annular	annular
5.0	annular	annular
10.0	annular	annular
17.0	dispersed	dispersed

3.4.3 Flow Instability

This problem can occur if the heat input in certain rows of collectors is reduced while others remain at normal heat-flux conditions. Because of lower heat flux, the pressure drop in those rows would decrease, resulting in greater flow through them and lesser flow through the other rows. Lower flow would generate greater steam quality which would increase pressure drop through them. The process could lead to a dry-out condition and render the system inoperative. Small differences in the heat flux of each row would not cause this problem. A constant-speed pump would increase the net flow rate as pressure drop through the collector decreases and a stable thermal-hydraulic configuration would be achieved. However, the problem could become serious if there was a cloud cover above a number of collector rows, cutting the heat flux in those collector rows; or the drive mechanism in a number of rows failed and the reflectors were thrown in the stow position with the fluid flow continuing.

Both of these conditions will be eliminated by using the proper operating procedure and an adequate control system. For example, if the drive mechanism fails in any reflector row, the system will not be operated unless the flow through that row is cut-off. One sun-sensor will be placed at each of the four corners of the collector field. If clouds cover any of the sun sensors, after a preset time delay, the solar system will be shut down until the clouds move away.

In summary, the flow instability problem is easily manageable.

3.4.4 Circulation Ratio

As the circulation ratio increases, the water flow rate in the loop increases, resulting in a higher system pressure drop and greater required pump power. The average temperature through the collector increases slightly and the steam quality at the collector outlet decreases as the circulation ratio is increased. A summary of computations for the range of circulation ratios between 10 and 50 is given in Table 3.8. Based on the discussion presented in Subsection 3.4.1, a circulation ratio of 20 may be selected. However, for the present design, a more conservative value of 25 was selected.

Table 3.8 Effect of Circulation Ratio on Collector Outlet Steam Quality

Circulation Ratio	Flow Rate (gal/min)	Average Temperature Through Collector (°F)	Steam Quality at Collector Outlet (%)
10	37	365	9.9
20	73	366	4.8
25	92	367	3.6
30	110	368	2.8
50	185	370	1.7

3.4.5 Receiver Tube Configuration

In contrast with System Alternates 1 and 2, System Alternate 3 is very sensitive to the receiver tube configuration. Three viable variations of the receiver configuration are shown in Figures 3.6 and 3.7. These are:

- Single-tube
- U-tube
- Double-tube.

Figure 3.6 Alternate Receiver Configurations

(A) SINGLE-TUBE

(B) U-TUBE

(C) DOUBLE-TUBE

Figure 3.7 Side View of the Receiver Configurations

The single-tube configuration utilizes a secondary reflector, thus a small amount of heat flux exists on the top surface of the tube. Therefore, the single-tube configuration is more prone to dry-out and corrosion. For a fixed value of circulation ratio and the number of collector rows, the mass flow rates through the single-tube and the U-tube configurations would be twice that of the double tube. Therefore, the collector pressure drop would be the lowest through the double-tube and the largest through the U-tube.

A cost-effectiveness comparison of the three configurations is presented in Table 3.9. An inlet pipe size of 2 in. O.D. and an outlet pipe size of 4 in. O.D. was used for this study. Based on these results the double-tube configuration was selected for this system.

Table 3.9 Cost-Effectiveness of the Three Receiver Tube Configurations

Receiver Tube Configuration	Piping and Receiver Tube Cost Difference From Base (\$)	Relative Pump Power	Unit Cost Per Net Thermal Energy Produced (\$/10 ⁶ Btu)
Single-tube	Base	1.0	141.76
U-tube	- 600	1.21	143.40
Double-tube	+1800	0.72	140.24

3.4.6 Number of Collector Rows

As the number of collector rows is increased, the fluid flow rate through each row decreases, resulting in a reduction in collector pressure drop and less

required pump power. Therefore, a larger number of collector rows would yield a more cost-effective system. However, as the mass velocity decreases, the probability of dry-out increases. In Subsection 3.4.1, the minimum safe value for mass velocity was established at $400,000 \text{ lb/h}\cdot\text{ft}^2$.

For a mass velocity of $400,000 \text{ lb/h}\cdot\text{ft}^2$, a maximum of five rows can be allowed in this system. Therefore, the collector field was optimally divided into five rows of collectors.

3.5.7 Collector Inlet and Outlet Pipe Size

Pipe sizes considered in this study ranged from 1-1/2 to 3-1/2 in. nominal diameter for the inlet piping and from 3 to 6 in. nominal diameter for the outlet piping. Larger sizes for the outlet pipe had to be considered because two-phase flow conditions require a greater pressure drop than single liquid phase flow. A summary of these computations for the inlet and outlet pipes is presented in Tables 3.10 and 3.11 respectively. Based on these results, the optimum size for the inlet pipe is 3 in. and 4 in. for the outlet pipe.

3.4.8 Optimized System Alternate 3

- Steam Drum
 - No steam dryer required.
- Piping
 - Inlet: 3 in. Sch. 40 insulated
 - Outlet: 4 in. Sch. 40 insulated
 - No heat tracing.

Table 3.10 Cost-Effectiveness vs. Inlet Pipe Size
(Outlet Pipe Nominal Diameter = 4 in.)

Nominal Diameter Inlet Pipe (in.)	Pipe Cost Difference From Base (\$)	Relative Pump Power	Unit Cost Per Net Thermal Energy Produced (\$/10 ⁶ Btu)
1-1/2	- 700	2.75	150.72
2	Base	1.0	139.40
2-1/2	+ 850	0.56	137.14
3	+1700	0.38	136.43
3-1/2	+3100	0.33	136.80

Table 3.11 Cost-Effectiveness vs. Outlet Pipe Size
(Inlet Pipe Nominal Diameter = 3 in.)

Nominal Diameter Outlet Pipe (in.)	Pipe Cost Difference From Base (\$)	Relative Pump Power	Unit Cost Per Net Thermal Energy Produced (\$/10 ⁶ Btu)
3	-1775	1.92	136.88
3-1/2	- 875	1.29	135.89
4	Base	1.0	135.66
5	+2800	0.74	136.36
6	+3825	0.66	136.65

- Pump
 - Centrifugal pump, single speed-motor
 - Motor size = 2 hp
 - Annual Pump Energy Consumed $\approx 12.2 \times 10^6$ Btu electric
- Receiver Tube Configuration
 - Double-tube
- Operating Conditions
 - Circulation ratio = 25
 - Fluid flow rate = 92 gal/min
 - Average temperature in collector = 367°F
 - Steam quality at collector outlet = 3.6 percent
 - Mass velocity through receiver = 434,780 lb/h·ft²
- Freeze Protection
 - Hot water circulation system
 - Existing feedwater pump utilized
 - Flow rate = 1.5 gal/min
 - Estimated annual energy consumed = 25.9×10^6 Btu
- Energy Production
 - Annual thermal energy produced = 2247×10^6 Btu

3.5 COMPARISON BETWEEN ALTERNATE SYSTEMS

A comparison of the cost-effectiveness of the three optimized system concepts is presented in Table 3.12. The cost of common items for the three concepts was assumed to be \$300,000. The installed cost of additional components was estimated on the basis of vendor data. These additional costs are provided in Table 3.12. It must be emphasized that the parasitic power has been computed in terms of equivalent thermal energy. Electrical parasitic energy has been multiplied by a factor of 3 for this conversion.

Table 3.12 Comparison of the Three Alternate System Concepts

Concept	Cost Difference From Reference of \$300,000 (\$)	Annual Energy Produced (10^6 Btu)	Parasitic Power* (10^6 Btu Thermal)	Cost Effectiveness [†] (\$/ 10^6 Btu)
System Alternate 1 (Oil)	+40,400	2093	32.1	165.17
System Alternate 2 (Flash Tank)	+31,000	2206	337.9	177.19
System Alternate 3 (Collector Boiling)	+35,600	2247	62.5	153.63

*Includes operation and freeze protection.

†Unit cost of net annual thermal energy produced.

The results presented in Table 3.12 show that System Alternate 3 is the most cost-effective concept. It is 7.5 percent more cost-effective than System Alternate 1 and about 15 percent more cost-effective than System Alternate 2. Therefore, System Alternate 3 was selected. The overall advantages of this system are:

- Most cost-effective, least expensive and yields highest thermal efficiency.
- Safest--eliminates hazards of organic oil (see Section 5).
- Simplest--requires least number of components.

- Has potential for use in the generation of higher temperature process steam. If a collector is useful up to 600°F, this concept would enable the collector to generate steam at 600°F whereas other concepts would be more limiting.

SECTION 4
ENVIRONMENTAL IMPACT ASSESSMENT

Section 4

ENVIRONMENTAL IMPACT ASSESSMENT

4.1 HISTORICAL

The proposed site was purchased by Dow Chemical, U.S.A., in 1966 from J. R. Proffit and C. Underwood and is wholly or partially contained in Land Lots No. 202 and 203, 13th District, 3rd Section, Whitfield County, Georgia.

No known archeological or historical sites will be involved in the proposed construction or operation.

4.2 TOPOGRAPHY AND SOILS

The site itself has previously been partially graded for planned industrial development. Additional changes to the existing site will be minimal. Some compacting of the soil is expected during construction and operation. This could result in clogging of the soil and additional rain runoff to the existing storm sewer system.

4.3 HYDROLOGY AND WATER QUALITY

The plant currently operates under a NPDES Permit No. GA0000426. The solar facility will not affect the conditions of this permit.

If the successful design uses aromatic hydrocarbons as the heat transfer medium, a diking system will be included. This system will assure collection of any fluid spills for proper disposal. Collected rain water will be released during dry periods. Surface water runoff will occur only during intense storms or with moderate rain on frozen grounds.

4.4 ODOR

No unusual or strong odors are anticipated from the construction or operation of the facility.

4.5 AIR QUALITY

Minor impact on air quality would occur during construction from emissions of construction vehicles and welding equipment, paint sprayers, and other equipment.

A possible source of air pollutants from operation would be release of the pressure relief system under emergency conditions. The probability of this occurrence is extremely small, however.

The Dow plant site operates under permit No. 2822-155-5961-0 and 2822-155-5965-0 from the Georgia Department of Natural Resources. All necessary permits for construction and operation of the proposed facility would be applied for before construction begins.

4.6 NOISE

Existing noise levels on the site are low. Plant operations would have no unusual noise factors. Minor noise would occur from diesel engines of construction vehicles.

4.7 VISUAL

There will be no significant visual impact resulting from this project. The height of buildings, tanks, and other structures is relatively low.

4.8 COMMUNITY SERVICES AND FACILITIES

The existing Dow plant will provide on-site fire protection facilities, and plant personnel receive training in fire prevention and suppression. This minimizes the possible impact on the adjacent communities' fire protection facilities.

Minor injuries to personnel will be handled through on-site first aid facilities. First aid training is given to plant personnel. More serious injuries will require treatment at Hamilton Memorial Hospital.

Security will be provided by plant personnel. The plant is operated on an around-the-clock basis.

4.9 WILDLIFE

No significant impact to terrestrial wildlife is anticipated. Negligible wildlife habitat would be lost as a result of the construction. There are no endangered species of animals or plants in the immediate area.

SECTION 5
PRELIMINARY SAFETY ANALYSIS

Section 5

PRELIMINARY SAFETY ANALYSIS

5.1 INTRODUCTION

The intent of this preliminary safety analysis is twofold:

- To utilize safety hazard data in selecting a conceptual design of a unit for solar production of industrial process heat at Dow Chemical Company's Dalton, Georgia plant
- To recommend features for inclusion in the detailed design and operating procedures of this unit.

In this preliminary analysis potential problems in the designs under consideration were identified. In this section, the relevant properties of possible working fluids are discussed, and features or procedures are recommended to enhance the safety of the unit should the design be adopted.

5.2 ALTERNATIVE CONCEPTUAL DESIGNS

Three alternative conceptual designs for the Dalton, Georgia unit have been described in Section 3 of this report. Briefly, they are:

- System Alternate 1. A system in which a heat transport loop delivers a hot organic fluid from the collectors to the boiler and recirculates the fluid to the collectors. Feedwater from the existing plant steam system is converted into saturated steam at 155 lb/in²g in the boiler.
- System Alternate 2. A system in which pressurized water is circulated through the collectors to a flash boiler where steam is generated. The feedwater is fed to the flash boiler. A level controller regulates the feedwater flow rate.
- System Alternate 3. A system in which water is partially boiled in the collector and circulated through a steam drum where the steam is separated. The feedwater flow rate is regulated by a level controller on the steam drum.

5.3 POTENTIAL SAFETY PROBLEMS

Five safety hazards associated with the system and remedial measures for correcting them are detailed in the following subsections.

5.3.1 Release of an Organic Heat Transfer Fluid to the Atmosphere

System Alternate 1 requires the use of an organic heat transfer fluid. The release of this fluid to the atmosphere creates the risk of fire and toxic damage to the environment.

The heat transfer fluids considered for use in this design are listed in Table 5.1 with their safety-related properties. Though none of the organic fluids under consideration pose major health hazards, ingestion will cause sickness, skin contact may cause irritation, and exposure to their vapors is irritating at levels of 10 ppm or higher.

The flash points of the organic heat transfer agents vary between 145°F for Dowtherm J to 425°F for Caloria HT-43. As the working temperature in the collectors is normally ~500°F, fire can result from spills if an ignition source is present, particularly with Dowtherm J. A water spray system covering the collectors is recommended.

Autoignition of the organic heat transfer agents is unlikely unless catalyzed by insulation. The likelihood of this can be diminished by the use of closed cell insulation and other measures.⁶

Table 5.1 Properties of Heat Transfer Fluids

Fluid	Dowtherm G	Dowtherm LF	Dowtherm J	Caloria HT-43	Therminol SS	Therminol 60	Therminol 66
Boiling point °F	700	600	575	700-1000	>610	600	644
Pour/freezing point °F	20	<-25	<-100	15	-40	-90	-18
Flash point °F	305*	260*	145*	425*	350*	310*	354 [†]
Autoignition Temperature °F	1030	1020	806	---	675	835	705
Degradation rate at 600°F (2 week)	0.011	0.011	0.450	2.50	N.A. [§]	N.A. [§]	0.185
Toxicity: skin contact	Mild irritant	Mild irritant	Mild irritant	Mild irritant	Mild irritant	Not a primary skin irritant	Mild irritant
8 h threshold limit value (TLV) ppm	N.A. [§]	~1 (based on smell)	10	N.A. [§]	N.A. [§]	N.A. [§]	N.A. [§]
Ingestion	Low toxicity	Low toxicity	Low toxicity	Low toxicity	Low toxicity	Low toxicity	Low toxicity
Composition	Mixture of di- and triaryl ethers	Alkylated diphenyl and diphenyl-oxide blend	Alkylated aromatic	Paraffins	Alkylated benzene	Aromatic hydrocarbons	Modified terphenyl

*Cleveland Open Cup (COP).

[†]Fensky-Marten.

[§]Not available.

5.3.2 Overpressurizing the System

Overpressurizing the system can result in damage to equipment and release of hot, possibly flammable fluids. One potential cause of overpressurization is the thermal expansion of liquid in an isolated solar collector. To guard against this, it is recommended that pressure relief valves be installed between the valves which isolate the collectors and that alarms or other devices be installed to warn the operators of the situation. These could take the form of low-flow alarms or limit switches and lights indicating valve positions. The relief valves should discharge to the dump tank if an organic heat transfer agent is used.

System Alternate 1, which utilizes an organic heat transfer fluid, includes an expansion tank to compensate for the expansion of the fluid as it is raised to the operating temperature. It is recommended that this expansion tank be of a size and that the liquid be maintained at such a level that overpressurizing the tank will not result if the tank cannot be vented as the organic liquid is heated and expands. In addition, a pressure relief valve should be placed on the expansion tank.

Water entering the hot organic liquid will vaporize and perhaps result in flow problems or increased pressure. Water may be introduced prior to start-up or may enter as a result of leaks in the boiler where the steam pressure greatly exceeds that of the organic liquids. Venting the expansion tank will remove this water from the system.

If water is to be circulated through the solar collectors, it must be prevented from freezing and expanding and rupturing the collectors. Because heat-tracing of the collectors is not feasible, draining the collectors or maintaining a steady flow of warmer liquid through them are procedures that will prevent rupture at freezing temperatures.

Similar measures should be adopted to prevent damage to the system should the organic fluid freeze and contract within the collectors. An organic heat transfer agent that is unlikely to freeze at the Dalton, Georgia location should be selected.

5.3.3 Overheating the Collectors

Should the flow of fluid through the collectors be reduced through partial blockage or be stopped by pump failure or valve closure, overheating can occur under high heat-flux conditions. As a result of overheating, thermal stresses can damage the equipment and, in the case of the organic liquids, degradation of the liquids can occur causing the formation of sluggish tars and even coke after long exposure.

The likelihood of the collectors overheating will be diminished by rotating the collectors to the stow position on failure of the pump, on closing of the valves isolating a collector, or when the flow through a collector is low.

5.3.4 Corrosion

Corrosion may be a problem with organic fluids if contaminated with water. Again, the importance of venting this water should be stressed. It

should also be noted that the organic heat transfer agents are incompatible with copper and aluminum at temperatures exceeding 400°F. These materials should not be utilized in the design.

5.3.5 Boiler Hazards

No exceptional hazards have been identified for the boilers in any of the designs.

5.4 CONCLUSIONS OF THE PRELIMINARY SAFETY ANALYSIS

Three main conclusions were drawn from the preliminary safety analysis:

- Associated with the use of an organic heat transfer agent is the potential for fire and toxic damage. This must be guarded against.
- A reliable means to prevent the overheating or overpressurizing of the system, and in particular the solar collectors, must be incorporated in the design or operation of the unit.
- Freezing of the process fluid that circulates through the solar collectors must be guarded against.

SECTION 6
REFERENCES

Section 6

REFERENCES

1. V. E. Dudley and R. M. Workhoven, "Summary Report: Concentrating Solar Collector Test Results," Sandia Laboratories SAND78-0815, May 1978.
2. V. E. Dudley and R. M. Workhoven, "Performance Testing of the Suntec SLATS Solar Collector," Sandia Laboratories SAND78-0623, 1978.
3. A. Pikulik and H. E. Diaz, "Cost Estimating for Major Process Equipment," Chemical Engineering, October 10, 1977, pp. 106-122.
4. J. G. Purnell and T. M. Novak, "Dry-Out and Pressure Drop Tests for Inclined Boiler Tubes," Report 3049, Naval Ship Research and Development Laboratory, Annapolis, Maryland, September 1969.
5. L. S. Tong, "Boiling Heat Transfer and Two-Phase Flow," John Wiley and Sons, Inc., New York, 1965, (Fig. 3-1, p. 49).
6. A. R. Albrecht and W. F. Seifert, "Accident Prevention in High Temperature Heat Transfer Fluid Systems," Loss Prevention Symposium, 67th Annual Meeting, Atlanta, Georgia, February 15-18, 1970.