SOLAR THERMAL ELECTRIC POWER SYSTEMS
WITH LINE-FOCUS COLLECTORS

Final Report

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EXECUTIVE SUMMARY

Objectives

Electric power generation by conventional Rankine cycle heat engines with heat supplied by line-focus solar collectors was investigated. The objectives of the study were:

1. Determine which of four types of line-focus solar collectors coupled with turbine-generators of conventional design has the potential to produce low-cost electric power with thermal energy in 100 to 300°C range.
2. Develop performance and cost relationships for organic Rankine cycle engines for power generation capacities from 3 MW_e to 300 MW_e.
3. Develop conceptual storage units for organic fluid systems.

Line Focus Collectors

Four types of line focus collectors were investigated in this study. They are:

- (1) Parabolic trough concentrating collector (PTC)
- (2) Rotating slit concentrators with fixed absorber (RSC)
- (3) Movable absorber concentrator with fixed reflectors (MAC)
- (4) Compound parabolic concentrating collectors (CPC).

The parabolic trough collectors are of conventional design but specific units of the types designed by Honeywell and Sandia (at Albuquerque, N.M.) were investigated to establish parametric cost models suitable for the computerized optimization process. Parabolic troughs in both east-west and north-south orientations were included in the optimization studies.

The rotating slats collectors involves narrow reflectors which are pivoted at the ends so that the reflecting surface can be oriented to reflect solar rays onto a fixed absorber. The reflections are generally on a plane and can be oriented with the mirror and absorber axis in a north-south direction and tilted with respect to horizontal, or aligned with the axis in an east-west direction with the mirror plane tilted toward the equator to increase the interception of solar rays with a fixed number of reflectors. Specific concentrators of this type investigated were designed by Sheldahl (Minnesota) and ITEX.

The movable absorber concentrator is a unique design that utilizes fixed reflectors oriented with the axes east-west and arranged in such a way as to focus the rays on a line regardless of the daily and hourly change in the position of the sun. The reflectors are placed on the circumference of a circle and the focal line moves along the same circumference. The absorber is moved with the changing focal line to intercept the concentrated solar radiation. A collector concept developed by General Atomics was investigated.

The compound parabolic concentrator was developed by Argonne and the University of Chicago, and utilizes multiple parabolic shaped reflectors in a plane to focus the solar rays. The design is unique in that both diffuse and direct radiation from the sun can be concentrated from a view angle that can be varied for specific designs. The concentration ratio can be varied from 1 to 10 and is suitable for generating temperatures in fluids to 300°C. The CPC Collector can be oriented in either north-south or east-west directions with the plane of concentration tilted to maximum advantage.

Rankine Cycle Engines with Organic Working Fluids

Selection of an organic fluid that could be suitable as a working fluid for Rankine cycle engines depends upon the characteristics of the turbine, condenser and feed heating systems, so that conceptual designs of the heat engine subsystem must be made to enable the selection of appropriate fluids. Turbine size, vapor moisture content, energy transport rate, condenser pipe size and feed-heat rates are the important design factors and the thermodynamic and physical properties of the fluid which are suitable for specific designs are also important.

From a large number of candidate organic fluids, consideration of critical temperature, vapor pressure and latent heat eliminated many classes of compounds. There was no single fluid which emerged as top choice for the entire temperature range from 100°C to 300°C. Ammonia appears to be a suitable fluid in the 100°C temperature region, and although ethyl mercaptan appears to be a suitable fluid in the 200°C region, not enough is known about this fluid so that it could not be recommended. Methanol is considered a better choice for this temperature because of cost as well as its properties. Pyridine and water in a 75/25 mixture by weight is the top choice for organic fluids in the 300°C region. Fluorinol-85 is also considered a possible candidate.

In comparison to steam, however, none of the fluids appear to have a clear economic and technical advantage for use in Rankine cycle engines. All fluids are expensive compared to water and thermal storage and heat exchange equipment are needed unless the organic fluid is circulated through the collector field. Circulating organic fluids through the collectors is not a preferred choice because of initial and operating costs with some degradation in collector performance because of poorer heat transfer properties compared to water.

Rankine cycle efficiencies using the chosen organic fluids and conceptual turbine designs are in the range of 9 percent for ammonia in the 80 to 130°C range, 12.5 to 15.5 percent efficiency for methanol from 130 to 225°C respectively and 29.5 to 33 percent for pyridine-water from 250° to 325°C. The estimated cost (1975 dollars) of the turbo-generator equipment is about \$725/kW for methanol and pyridine/water engines and about \$375/kW for an ammonia engine. Condenser, piping, heat exchanger, cooling and electrical plant equipment costs add several hundred dollars per kW to these costs, exclusive of the solar collectors, and piping for the collectors. The cost/kW_e depends on plant capacity and costing relationships are developed in the report.

Optimized Systems

Parabolic trough concentrating collectors in distributed systems provide the lowest production cost for electrical energy from solar thermal electric power systems using line-focus collectors. Production costs (exclusive of operating and maintenance costs) are projected to be about 6.5 ¢/kWh from PTC systems, 8.5¢/kWh from RSC systems, 9.7¢/kWh from MAC systems and 31.7¢/kWh from CPC systems. Operating temperature which provides lowest cost energy for the PTC, RSC and MAC systems is 250°C, although the cost for electrical energy was identical for the PTC system at 300°C. The best operating temperature for the CPC system is 150°C.

In all collector systems, the preferred alignment is the east-west direction when optimized in the collector field because of optical considerations and the absorbers become part of the fluid circulation pipe network in the field. Collectors should be tilted toward the south to maximize aperture width, and fixed tilts are more economical

than providing adjustable tilts to the large collectors. The structure and adjusting mechanisms for adjustable tilt collectors were found to be expensive, especially to withstand large wind loads.

Conclusions

Parabolic trough collector systems have the potential to produce lowest cost electrical energy from solar thermal power plants which utilize heat from line-focus collectors. The preferred operating temperature in the collector field is in the range from 250 to 300°C. The cost of electric energy is relatively insensitive to plant capacity, but plant sizes of about 30 kW_e appears best. However the difference in cost of energy produced from smaller plants of 3 MW_e size is not significant considering the approximate nature of the costing procedure. Although electrical energy production costs may not be certain, the comparison between different collector systems is valid in this study because the performance and cost models are consistent and were developed without advocating a particular system.

The preferred heat transfer fluid in the collector is steam, because of transport costs through the collector field, and the advantage of cooler return temperatures to the collectors from the condenser. Water is boiled in the absorber and is transported to the turbines (and storage) as steam.

There are three organic fluids suitable as working fluids in Rankine cycle engines, each appropriate for a specific temperature range. At 100°C ammonia is preferred, at 200°C methanol is suitable and at 300°C, pyridine/water mixture appears best. The thermal efficiency is acceptable for the organic fluid engines, but the cost for the turbine generator subsystem is also more expensive than steam systems

in the 250 to 300°C range. While the cost of power generation hardware for ammonia systems is comparable to that for steam systems, low temperature working fluids in the 100°C region result in higher electrical energy cost compared to higher temperature systems. Thus, steam systems at 250 to 300°C range resulted in lowest cost electrical energy production, with parabolic trough collectors in the field producing heat directly in steam at the absorber.

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1.0 INTRODUCTION

1.1 GENERAL CONSIDERATIONS

The minimum cost of electric energy from solar thermal electric power systems (STEPS) with distributed collectors were determined for four different collector types. The collectors analyzed were (1) parabolic troughs (PTC). (2) Fixed absorber with rotating slats (RSC). (3) Fixed slats with a rotating absorber (MAC). (4) Compound parabolic concentrator (CPC).

The parabolic trough depicted in Figure 1-1 is tracked on one axis so that the plane passing through the axis of symmetry of the parabola always includes the sun. The RSC collector shown in Figure 1-2 is positioned at a fixed slope and utilizes movement of slats to redirect the sun's rays to a fixed absorber. The MAC collector shown in Figure 1-3 uses a moveable absorber and fixed geometric orientation of the slats to direct the sun's rays to a common focal line coinciding with the absorber. In this study the CPC concentrator shown in Figure 1-4 is positioned at a fixed slope. The sun's rays are concentrated on the absorber from parabolic shaped sides of the trough section. If desired, the CPC may be moved to track the sun.

A significant amount of effort was devoted in this study to develop parametric cost and performance models for each of the four types of collectors. The performance and parametric cost models were derived so as to be suitable for optimization.

The parametric cost models express subsystem costs as functions of the design parameters. For example the models yield the cost of a parabolic trough with a 35 degree rim angle as well as one with an

Figure 1-1. Parabolic Trough Collector (PTC)

Figure 1-2. Linear Array of Rotating Slats with a Fixed Absorber (RSC)

Figure 1-3. Circular Array of Fixed Slats with a Movable Absorber (MAC)

Figure 1-4. Compound Parabolic Concentration (CPC)

80 degree rim angle. The cost estimation effort is uniform in that all cost models are developed by using the same methodology. This feature of the study allows cost comparisons to be made objectively between the different types of collectors.

Performance models for the collectors were also derived in a form that is suitable for the optimization procedure. Simple and accurate analytic models have resulted from this study.

A dynamic programming systems optimization approach was used to determine minimum-cost solar thermal electric power systems. Dynamic programming requires the selection of appropriate performance parameters to concisely and completely represent the performance of one subsystem as it affects the next subsystem. For example, the concentrator effect on absorbers and collector effects on piping and piping layout, are parameterized.

The optimization procedure used in this study assures an optimum system and is organized so that the system performance models are essentially the same for all four types of collectors from the absorber through to the turbine-generator-cooling tower subsystem.

The analytical concentrator models that characterize the concentration of radiation for each of the four types of collectors are substantially different. The four different performance models represent a major research effort and, it is believed to be an extension to the state-of-the-art in collector analysis.

The absorber used for three of the four collector types is within an evacuated tube and coated with a selective surface. The absorber tube is insulated on the back side to further reduce thermal losses. In the CPC the absorber tube is replaced by a fin-tube arrangement. The

actual sizes of the absorbers which evolve in the optimization program are different. For example, the width of the absorber is greater for concentrators where the spread of radiation at the focal zone is wider, as with the RSC and MAC concentrators.

1.2 CONCENTRATOR MODEL DIFFERENCES

The differences in the design and operation of the four types of concentrating collectors considered in this study required different models. However, a common form of analysis is required to make the comparisons valid. Design variables common to all four line-focus concentrators of this study are aperture width, pointing error, surface contour error, surface reflectance, and rim angle. For the CPC the acceptance angle is used instead of the rim angle. Design variables that are common to some of the collectors but not to others are: (1) slope for the RSC, MAC and CPC collectors; (2) slat width for the RSC and MAC collectors; (3) tangent slat location for the MAC collector; and (4) number of CPC units in a box and trough height for the CPC collector. The trough height is a variable which relates to the degree of truncation from a full CPC.

In contrast to the other three types, the CPC is a non-imaging device. Analysis of this collector therefore required modification from the image spread measure of the concentrator performance characterization.

Dynamic effects are important in the analysis and are accounted for in the performance models of the four concentrators of this study. These effects include considerations of the time of day and year to compute sun position, variation in levels of insolation and ambient temperature. The inclusion of dynamic effects are required to properly

assess the effects of blocking and shading, cosine angle, end spill, and image spread as a function of sun position. Thermal losses are greater for line-focus devices than for point-focus devices and therefore affect performance at different insolation levels. Also, because the absorber area of line-focus devices is greater, ambient temperature variation has a more pronounced effect.

Image spread is affected by a number of factors. Basic among them are inherent inaccuracies in tracking and irregularities of the reflective surface. Also considered in the image-spread model is the defocusing by the slot type collectors. The curvature of the RSC slats also affects the image spread as the sun position varies. All of the collectors exhibit an additional spread of the image for non-perpendicular radiation. This is due to the lack of a focal point for an individual slot with non-normal radiation. In the MAC collector the sun position affects the absorber position which also affects the distance a ray travels to reach the absorber. The collection of diffuse radiation by the CPC is accounted for in the model.

1.3 COMPARISONS

Comparisons are made between systems delivering thermal energy at temperatures of 100, 150, 200, 250, and 300°C. The systems analyzed all produce saturated steam in the collectors because boiling systems perform better than pressurized hot water systems at these temperatures [1]. This is mainly due to heat losses in a boiling system through the return line to the field because the return temperatures to the collectors are at the condenser temperature which, for steam systems, is about 40°C as compared to a hot water system where the return temperatures are usually within 50 degrees of collector outlet temperature.

The need for an elevated return temperature is related to the pinch-point characteristics in the boiling heat exchanger. The pump capacity for the hot water system must be greater than for steam to transport heat at the same rate, hence the pumping power required is greater for the hot water system.

Comparisons are made for a fixed set of collector and absorber characteristics. The absorptance was set at 0.95, the emittance at 0.28, and the transmittance at .95. This corresponds to a selective surface that is readily achievable. The reason for fixing some variables in the analysis is to limit the computer time (and funds) in the study.

2.0 COST MODELING OF COLLECTORS

2.1 INTRODUCTION

This section covers the procedures and results of the costing effort for the four types of concentrating collectors. The optimization model relies upon the cost-design relationships for these collectors. The search for cost-effective designs resulted in the selection of those manufacturing, installation, and other factors that were found to be cost dominated. That is, for a given set of performance characteristics, least cost alternatives have been chosen. Where trade-offs exist between cost and efficiency, then these sets of values were conveyed to the optimization model. While a number of these trade-offs have been reasonably well established, much remains to be done in providing a more complete set of values that would adequately utilize the power of the optimization techniques.

The following sections describe the scope and methods of cost modeling, various design factors used in estimating the costs and results for a standard absorber design. Costs for four different concentrator designs are detailed in appropriate sections covering each type of concentrating collector.

2.1.1 Scope of the Analysis

The four collector types analyzed were: the parabolic trough collector (PTC) as designed by Honeywell and Sandia, the fixed absorber with rotating slats (RSC) as designed by ITEK, the compound parabolic concentrator (CPC) as conceptualized and developed by the Argonne National Laboratories, and the fixed slats with movable absorber (MAC) as conceptualized by General Atomic. Performance parameters for which cost responses were sought are aperture width and area, rim angle,

reflectivity, contour and tracking accuracies, collector orientation, and fluid pressures and temperatures in the absorber.

Ideally, the costs associated with each of the ranges of parametric values would include annualized values for the costs of investment, replacement, maintenance, repair, and system operations. The absence of reliable life and operational estimates, however, together with the predominance of investment costs have led to the use of investment costs alone for the cost models. This approach in costing tacitly assumes that the costs of replacement, maintenance, repair and operation are either similar among the various designs of the four collector types or that the costs of these items relative to investment costs is not significant. Reports by Honeywell (1) (2) suggest that the durability of the reflective surfaces of the collectors and the coatings for the absorbers may be quite adequate for practical purposes. Assuming this is correct, the present value of future replacement costs would be small. After 58 weeks of testing, the reflective surfaces of Teflon and glass have shown no appreciable deterioration, and absorber coating life can be assumed to be of the order of 20 years. Moreover, the concentrators can be designed so that reflector surfaces are replaceable. Honeywell replaced the surface of the parabolic trough test model in Arizona (from an original test material to one of Alzak) without difficulty or loss in contour accuracy. In an earlier study by Colorado State University and Westinghouse (3), the comparison between the costs of cleaning the reflective surfaces and adding collectors to make up the energy lost by reduced reflectivities of surfaces that are not cleaned, favored the alternative of not cleaning the collector surfaces. The costs of

effective cleaning methods are uneconomical. Further comments on replacement and annual costs will be made in the appropriate sections on individual collectors.

Other cost items omitted from this section on cost modeling are those common to all designs. These include land purchase and clearing, service roads, and engineering design. The costs of other subsystems such as pipelines and the turbine-generator-cooling units are included as parts of those subsystems.

2.1.2 Approach

As in the earlier STEPS study by Colorado State University and Westinghouse (3), cost modeling was carried out jointly between the staffs of CSU and Westinghouse Production Technology Center. Westinghouse provided the expertise in designing and estimating the costs of new types of equipment and the University developed the cost models and estimated structural, construction and related costs.

Visits were made by the staff to each of the five organizations whose collector designs were studied. In some cases test models were observed, and in all cases discussions were held with those having responsibility for collector design and testing. Published reports, letters, and working drawings were also obtained. These data centered more on design, manufacturing, construction, and testing than on costing. Many of the cost estimates were not suited for the present study because they generally pertained to small-scale production of units that were experimentally designed for testing rather than for mass production. As a result, costs estimated by these organizations tend to be higher than those estimated in this study.

In the earlier CSU/Westinghouse study, large-scale production was assumed for establishing the manufacturing procedures and subsequently estimating production, construction, and installation costs. Large-scale production quantities were determined by assuming that five percent of the projected average annual additions to utility generating capacity for the 1970 to 1980 period in the United States would be supplied by solar energy power plants. Five percent was considered to represent a significant and realistic proportion of the total national power production capacity. A further assumption was made that the solar collectors would be manufactured by five plants strategically located across the United States and each plant could provide an annual output equal to one percent of the projected installations. In the present study, this annual capacity was reduced to 0.5 percent of annual output because the initial production capability would likely be low, building up to larger scale production after the first few years. The annual output per manufacturing plant is $7,000 \text{ m}^2$ of collectors per day, which is still sufficiently large to obtain significant economies of large-scale production.

The earlier STEPS study relied on "conceptual" designs in estimating the relative costs of alternative collector designs. In the present study, preliminary designs and drawings of collectors, were made for the purpose of cost estimations. The drawings reflect the essential detail features of the collectors being studied or contemplated by the independent designers, but design changes were made by CSU and Westinghouse with the changes resulting in lower manufacturing costs. While the designs of the various collectors are believed to lead to reasonable approaches to large-scale production and costs (in 1975 values), there are refinements that probably can be made to further reduce the cost of each of the five collectors analyzed in this study.

2.2 SOME DESIGN CONSIDERATIONS

The cost models are influenced by the methods assumed for manufacture, installation, and operation of the collector units. Some of the more relevant factors are discussed below.

2.2.1 Processes and Materials

Standard equipment and processes were selected as the basis for estimating the costs of manufacture. Whenever possible, advantage was taken of supplier discounts for large volume orders in standard sizes for items such as steel sheets and catalog items such as motor and gear reducers. In-house manufacture was assumed for most fabricated parts, because the annual volume of plant output is sufficient to gain economies of scale and thereby reduce costs relative to outside purchase. Assembly-line equipment would be used, involving processes such as automatic coil handlers, squeeze rolls, and curing ovens.

2.2.2 Absorber Movement

Difficulties were encountered in deciding upon the designs for certain absorber connections to allow for rotation and temperature changes. These difficulties are associated with the alternatives involving high pressures and large diameters for the parabolic troughs and the movable-slat collector. Data were not readily available for the lifetimes and costs of flexible couplings in diameters greater than 2.54 cm and pressures in the 90 kg/cm^2 (1300 psi) range. The approach taken was to select parabolic trough designs that did not involve rotating absorbers, as the review of the Westinghouse and Sandia collector designs showed little cost savings for the reflector that made use of a movable absorber. By removing the requirements of rotating the absorber, the only provision

for absorber movement was that associated with contraction and expansion due to temperature changes. Flexible connections are one possibility; another possibility is to use swivel joints. The latter was favored because of the uncertainty of the life and strength of flexible hoses at high pressures and diameters. In estimating the costs of the swivel joints, V-shaped joints were assumed to be placed every 100 m to allow for linear movements of approximately 50 cm over a temperature change ranging from 0 to 300°C. Three swivel joints would be employed, one at each point on the V, as shown in Figure 2-1.

Figure 2-1. Swivel Joint for Linear Absorbers

Because cost data could not be obtained directly for such swivel joints, costs for globe valves for the various sizes and pressures were used as surrogates. The cost estimates for the absorber designs include installation, supports, and insulation. These values are incorporated in the general cost model for the absorber.

Expansion and contraction for the CPC and fixed slat collectors were not thought to be a design or costing problem. While the absorbers of 7.5 to 15 m length for the fixed slat collectors could have corresponding longitudinal movements of 3.8 cm and 7.6 cm over a temperature change of 300°C, the rotational arms of 6 and 12 m length would result in a potential angular change of only 11 minutes at the points of connection.

Such a small angular change should be readily taken up by the flexure of the rotational arms as seen in Figure 2-2.

Figure 2-2. Rotating Absorber Arrangement

The serpentine shape of CPC absorber is of the form that naturally allows for expansion and contraction. Thus, the problem of absorber movement should not exist for this design.

2.3 Widths and Lengths

Different collector widths and lengths can be manufactured with essentially the same equipment and manpower. Of course dimensions and wind and dead loads are interrelated. Cost-width relationships have been parameterized, while cost-length relationships have not (except for one alternative for the CPC collector). The approach taken for the lengths of manufactured items was to assume a convenient length for manufacture and transport that would not give excessive errors due to deflection (e.g., 1 cm in 6 m for the movable slats collector). This approach to collector lengths assumes no great variance in costs with lengths of sections different from those selected for study.

2.2.4 Supports and Footings

Footings and supports were designed structurally for winds of 45 meters per second (100 mph) and operationally for winds of 14 meters per

second (30 mph). With fail-safe devices or advanced knowledge of heavy winds so that stowing could take place, structural damage is not expected even where such a wind velocity is exceeded. Wind loading becomes a major factor in the structural design of only the parabolic troughs. For the other collectors, wind loading is not great due to shape (CPC), small size (RSC), and nature of construction (MAC). Wind loading on the absorber is of negligible importance.

Alternative designs were considered for collector footings. The design selected for parametric costing assumes cylindrical footings of poured concrete with anchor bolts and base plates. The footings would be mechanically augered to a depth sufficient to guard against frost. The critical design factor for the troughs is failure of the footing in uplift. Pullout resistance due to skin friction was assumed for combinations of sandy and shale soils.

2.2.5 Collector Orientation

For horizontally mounted collectors, costs would be independent of orientation. For polar mounts (i.e., collectors tilted toward the equator) installation costs vary with the north-south length, since the volume of structural steel or earthwork increases with this length. For the troughs, the collector field was assumed to be shaped so as to give cross-sectional slopes of approximately 30 degrees. Large-scale earth moving equipment is able to form the desired inclination at a lower cost per unit area of collector than fabricated structural steel. Alternative detail structural designs, however, have not been carried out for all possibilities. For the rotating slats and CPC installations, the cost advantages of shaping the earth or building structural supports depend on the length in the north-south direction. For shorter lengths

structural supports appear more cost-effective; whereas for longer lengths, earthwork is probably the lower cost alternative. Finally, for the fixed-slat collector, the assumption of low-cost earth construction is fundamental to the design.

2.2.6 Reflective Surfaces

Reflective surfaces of back-silvered glass has been assumed for the parabolic trough and rotating slat collectors. Standard manufacturing processes can be used to mold the glass to the required shapes at low costs. Breakage is minimized by backing the glass surface with structural honeycomb and aluminized steel sections and assembling the units in the factory. Considering the favorable and known properties of back-silvered glass, other reflective surfaces were not evaluated for these two collector types. Alzak aluminum was considered for the CPC collector in accordance with the design considered by Chamberlain Manufacturing. The choice of aluminum as reflective material for this type of collector agrees with the findings of the first STEPS study. Aluminized Teflon or Mylar thin films are suggested for the fixed slat collector. Flexibility and resistance to breakage are considered important for this type of structure.

2.2.7 Reflectivity

Studies by Honeywell (1), Sandia (4), and Argonne Laboratories (5) indicate that initial specular reflectives for back-silvered glass might reasonably be about 85 percent, 77 percent for Alzak, and 80 percent for aluminized Teflon. On the assumption that cleaning would occur only for abnormally dirty conditions, the effective values over time could probably be 75 percent for back-silvered glass, 67 percent for Alzak, and 65 percent for aluminized Teflon.

2.2.8 Contour Error

This study has not been able to produce any further insight into the relationships between costs and contour error beyond that which has been reported elsewhere. The Honeywell study (1) reports a contour error of $1/2$ degree which, for their test model, gave a 4 percent reduction in radiation reaching the absorber area. The earlier STEPS study indicated that standard manufacturing processes could obtain accuracies within 0.1 degree and that little savings in cost would result from relaxing these tolerances. However, other factors also affect contour accuracies such as the accuracy of framing, dead and wind loading, and the "cording" effect when two surfaces are attached to each other. Sandia and others have suggested that such "cording" can be greatly reduced by end-clamping the sections and allowing the flexible reflector surface to take the shape of the supporting material.

With the parabolic trough as a standard giving contour accuracies on the order of 96 percent, relative accuracies of the other collectors can be compared qualitatively. The CPC collector is built in a manner not unlike that of the trough and thus might be expected to have contour accuracies of the same magnitude. The rotating slat collector makes use of long, thin slats that are more subject to deflections due to wind loading and vibrations than the troughs and so would be expected to be somewhat less accurate. Contour accuracies of the fixed slat design would tend to be the least accurate due to the difficulties of accurately positioning the reflective surfaces on concrete surfaces under field conditions, or of setting precast sections with the reflective surfaces already installed, if this method of installation is to be used.

2.2.9 Tracking Error

Tracking equipment with standard design can provide accuracies of 0.1 degree without significantly increasing design costs. Tests of the parabolic trough unit by Honeywell(1) has verified that such accuracies are readily achievable. At the same time, tests also show that tracking errors of up to 0.5 percent have little influence on collector efficiency. Given these conditions, an accuracy of 0.1 degree is assumed for tracking error for all collectors.

2.3 RESULTS FOR A SINGLE ABSORBER DESIGN -

A single basic design was selected for cost parameterization for the absorbers used with the parabolic troughs, rotating slats and fixed slat collectors. A sketch of one of the end sections is shown below and the material list can be found in the itemized calculations by Westinghouse for these collectors in Appendix A. This section describes the characteristics of this design and the variances in costs as a function of absorber pressure and diameter.

Figure 2-3. End Section Details for a Linear Absorber

The absorber used in the CPC collector is a serpentine steel tube, with a selective surface coating, which is encased in the collector module. The itemized absorber materials for the CPC absorber can be found in the Westinghouse calculations, also in Appendix A.

2.3.1 Absorber Description

The sketch of the absorber shown in Figure 2-3 depicts the essential features of the absorber used in calculating costs for the PTC, RSC, and MAC collectors. This absorber contains features that are generally consistent with the absorbers being considered by those experimenting with linear collectors: namely, glass encased carbon steel tube with the annulus evacuated at the factory, a selective coating, and bellows to take up the differential expansion. Detailed cost estimates of the troughs by Westinghouse failed to reveal any important savings in the collector shell and drives by going to a movable absorber instead of a fixed absorber. Considering the potential difficulties and costs associated with the movable absorber, all the calculations are based on the assumption that the absorbers for the trough are fixed. Of course the RSC collector is not confronted with this problem and in the MAC collector, the absorber must rotate.

2.3.2 Cost Formula

The cost formula below gives the costs per unit area of collector as a function of pressures and absorber diameters. With these two variables (p and d_s), the collector dimensions for aperture width and projected area (w and m), and by knowing the specific weights and unit costs of steel and glass (S_s , S_g , C_s , C_g), the remaining variables for pressure $f(p, d_s)$, steel and glass pipe thicknesses (t_s and t_g), and

diameter of the glass pipe (d_g) can be derived and the unit costs per square meter of aperture area can be calculated.

$$C = \frac{1}{w} [380 t_s S_s C_s (d_s - t_s) + 400 t_g S_g C_g (d_g - t_g) + f(p, d_s)] + \frac{1}{m} [0.19(d_g)^{1.5} + 0.48 d_g] + \left(\frac{2.00}{w} + 0.10 d_g\right) \quad (2-1)$$

where:

C is the installed cost of absorber in 1975 dollars per square meter of projected area

s and g are subscripts for steel and glass pipe, respectively

t is the thickness in centimeters

S is the specific weight in kg/cm^3

$$S_s = 0.00785$$

$$S_g = 0.00270$$

C is the unit cost in dollars per kg to the factory

$$C_s = 1.10$$

$$C_g = 1.76$$

d is the diameter in centimeters

w is the aperture width in meters

m is the projected area of reflector in m^2

$$d_g = 2 d_s \quad (2-2)$$

$$t_g = 0.016 d_g \quad (2-3)$$

$$t_s = 1.1 \left[\frac{d_s}{2} \left(\frac{\frac{p}{986}}{\frac{p}{986} + 1} \right) + 0.0135 d_s \right] \quad (2-4)$$

p is pressure in kg/cm^2

Values for movable connections, $f(p, d_s)$, for three pressures (p) and two ranges of pipe diameter (d_s) are given below. As noted above this function assumes these connections are spaced every 100 meters.

Values for $f(p, d_s)$
{1975 dollars per meter of collector length}

Pressure (p)	Absorber diameter (d_s)	
	$d \leq 4$ cm	$d > 4$ cm
16 kgs/cm ²	$1.40 d_s^{0.45}$	$0.27 d_s^{1.59}$
40 kgs/cm ²	$1.70 d_s^{0.55}$	$0.34 d_s^{1.63}$
90 kgs/cm ²	$2.07 d_s^{0.53}$	$0.38 d_s^{1.75}$

The $f(p, d_s)$ term represents the equations for movable couplings. The first of the next two terms contained in the brackets represents the cost of metal to glass seals and the expansion bellows. The second of these two terms is for miscellaneous hardware. The remaining two terms combine the costs of installation and selective coating materials.

Equation (2-2) shows that the diameter of the glass tube is twice the diameter of the steel pipe. For smaller diameter pipe, this relationship is close to that suggested as being optimum by Sandia (4). In this report by Sandia (page 31) an annulus distance between the two surfaces of about 1 cm was suggested as optimum when considering the combined effects of radiative, conductive, and convective heat losses. Thus, for an absorber tube with diameter of 2.0 cm, the diameter of the glass pipe would be 2.0 cm and resulting the annulus of 1.0 cm would equal the optimum. But as the radius of the absorber expands, the annulus gradually exceeds this optimum. Countering this optimization factor is

the use of the glass as a supporting structure for the liquid-filled steel tube. As the size of the absorber increases, its weight increases due to both the steel material and the liquid it contains; and the rigidity of the glass pipe is enhanced by its larger (than 1 cm) radius. It should be noted, however, that Equation 2-2 is a compromised rather than an optimized relationship.

Equation (2-3) was derived from the collapsing pressure for glass tubes according to the formula by A.E.H. Love. Equation (2-4) is taken directly from the ASME Boiler Code for carbon steel tubing. Equation (2-5) is the result of curves fitted to cost estimates for movable connections.

2.3.3 Absorber Costs

Absorber costs for specific collector designs are discussed in subsequent sections on overall costs of these collectors. Values for absorbers shown in these tables range from \$5.00 per square meter of aperture area for a 2.5 cm absorber mounted on a 2.13 m width parabolic trough to \$34.40 per square meter of aperture area for a 50.8 cm absorber mounted on a 24.4 m fixed slat collector.

The family of curves for alternative pressures and diameters for the 2.44 m x 3.66 m Sandia trough is shown in Figure 2-4. Increases in diameter have the most pronounced effect on absorber costs, with increased pressures contributing only a few dollars per aperture area in the smaller diameters. But at the larger diameters, increased pressure raises costs considerably. The primary cause of these dramatic increases in costs at the higher diameters stems from the assumed cost functions for the movable V-shaped joints. Since some absorber diameters will be in excess of 3 cm, then more thorough study of the cost of these couplings is warranted.

Figure 2-4. Installed Absorber Cost for 2.13 m x 3.66 m Sandia Trough for Various Absorber Diameters and Pressures

Absorber diameters shown in the Honeywell, and ITEK reports are only 1.9 cm to 2.5 cm.

2.3.4 Concentration Costs

Discussion of concentration costs and cost models are included in specific sections of the report for the different collector types. Because the objective of this study is to determine the optimum system which can produce electricity at lowest cost, cost models are important parameters in the optimization process. Furthermore, cost models which are parameterized with variables of collector design are needed to enable selection of optimum systems.

3.0 PARABOLIC TROUGH SYSTEMS

3.1 CONCENTRATOR PERFORMANCE MODEL

Analytical models (as differing from ray-trace models) were developed to represent concentrator performance in this study because of the need for rapid computation of the dynamic performance of countless thousands of concentrator design iterations in the optimization process. The performance models used in this study are largely extensions of models developed in the previous CSU/Westinghouse STEPS Study (3). The collectors in the present study are all line-focus collectors, as delineated in the scope of the project, even though point-focus collectors were found to be optimum for electric power generation in the previous study. With a point-focus device there are some simplifications in the analysis because the aperture with proper tracking, is always normal to the sun's rays. For a line-focus device, however, complications arise in the formulation of analytical models because the aperture is not always normal to the solar beam and cause such effects as variation in image spread, spills and other dynamic effects.

The basic approach for performance modeling initiated in the previous STEPS study enabled characterization of collection performance with two parameters. One is a measure of the capability of the concentrator to deliver radiation to the focal zone, the effective aperture, E , and the other is a measure of spread of the image at the focal zone, the spread distribution parameter, g . This basic approach is extended in this study to include the different positions of the sun relative to the reflector surface and the dynamic effects on the absorption for various types of concentrators.

3.1.1 Design Variables and Their Effects

The rim angle, $\theta_{\max}$, of a parabolic trough is the half angle from the axis of symmetry of the parabola to the rim of the trough measured from the focal point. The rim angle and imperfections of the reflecting surface are primary factors in determining the spread of radiation at the focus. Normally the image spread increases directly with increasing rim angles.

Collector aperture, A_p , is the area of an imaginary plane from rim to rim across the concentrator through which solar radiation passes before being concentrated. The aperture of a concentrator defines the absolute upper-bound for the amount of radiation that can be delivered to the absorber-heat exchanger. The aperture area also determines the image spread. For example, a concentrator with a one-meter aperture width may have an image spread of one centimeter width. If this same concentrator were scaled up to 10 meters, the image spread would be 10 centimeters. The size of the aperture therefore affects absorber size and cost. An absorber-heat exchanger that is sized for a 10-cm image spread will not generally have a 10 to 1 cost and performance difference over one that is sized for the 1-cm spread.

Any reflective surface or lense will have surface irregularities and will be subject to errors in tracking which differ from theoretically ideal conditions. These contour and pointing errors, labelled σ_ϕ and σ_δ , respectively, can have significant effect on the capability of the concentrator to deliver high temperature energy. The greater the surface contour errors and pointing error, the greater will be the image spread and therefore the larger will be the area of absorber-heat exchanger required to intercept the desired portion of the

concentrated radiation. With larger widths of absorber-heat exchangers, heat losses increase and costs are also greater. One of the unique features of the analytical performance models developed in these studies, is the explicit consideration to the effects of surface contour error and other factors on the spread of the sun's image on the absorber.

Specular reflectance of the surface, ρ_{ave} (or in the case of lenses, transmittance), is also an important consideration in collector performance. A surface with low reflectance reduces collector efficiency directly, and while a surface with high reflectivity is therefore desirable to achieve high collector efficiency, the cost of achieving such a surface may be uneconomical in terms of the cost of energy delivered from the subsystem. While there were many types of surfaces considered, based upon practical considerations of manufacture processes and durability, back silvered glass with a reflectance degraded to 0.78, to account for dust buildup, was used in this study. The significant design variables and their effects on the total performance of the concentrator are explicitly included in the analytical models.

3.1.2 Dynamic Effects

A parabolic trough is tracked on one axis so that the plane of symmetry of the paraboloid always passes through the sun. For an east-west orientation of the parabolic trough, the sun's rays are directly perpendicular to the aperture of the trough only at solar noon of each day. For a north-south oriented collector with a fixed tilt, the sun's ray is only normal to the aperture of the trough on the equinox days.

There are significant cosine effects that must be considered in modelling a parabolic trough. These are due to the angle of incidence with respect to the reflector surface as illustrated in Figure 3-1. The amount of direct normal incident radiation that can be intercepted by the aperture of the collector is reduced by the cosine of the angle of the sun's ray with respect to the aperture of the collector. The incident angle will depend on the collector orientation and the time of day.

Two types of blocking and shading effects are important for collectors. There can be blocking and shading by adjacent reflective facets and shading of the reflective surface by the absorber. For a parabolic trough, the only effect is shading of the reflective surface by the absorber. Some of the radiation which is blocked from the reflector is absorbed however.

End-spill occurs in all line-focus concentrators. For short-length collectors end-spill effects can significantly reduce the amount of energy collected. With long-length collectors the end-spill can be less significant.

Figure 3-1. Lengthening Effects of Solar Rays after Reflection

The spreading of the sun's image is affected by the incident angle. As shown in Figure 3-1, the path length of the reflected ray is considerably longer than if the incident sun's ray is directly normal to the reflection surface. Thus in the early morning or late afternoon the width of the solar image is considerably greater than at noon for an east-west oriented trough. If the absorber-heat exchanger is sized to intercept the reflected radiation for mid-day conditions than at other times of the day, not all of the reflected radiation will be intercepted. On the other hand, if an absorber is sized to intercept all of the reflected radiation all of the time, the absorber width will be large, which will result in greater cost, more blocking and shading of the reflector surface, lower concentration ratio and lower fluid temperatures. These effects are even more pronounced for the other types of linear collectors. Thus, the capability to treat image spread dynamically is an important consideration in performance modelling.

3.1.3 Analytical Models

For direct normal insolation the capability to deliver radiation to the focal zone, or effective aperture, of the parabolic trough is the product of the aperture of the trough and the average reflectance of the concentrator surface. This quantity multiplied by the incident radiation flux through the aperture is the amount of radiation delivered to the focal zone. Since the radiation flux through the aperture varies with the incident angle, the effective aperture also depends on the time of day and the season of the year. For example, for an east-west oriented parabolic trough, the effective aperture will be much less in the early morning than at noon on the same day.

If the surface contour and pointing errors are normally distributed and the radiation across the solar disc is uniformly distributed, the distribution of radiation at the focal zone is normally distributed with zero mean at the absorber diameter (3). This applies to all line-focus collectors including those of this study, regardless of shape. For a parabolic trough the standard deviation of this distribution, g , can be determined from Equation (3-1).

$$\frac{g}{A_p \sqrt{4\sigma_\phi^2 + \sigma_\psi^2}} = \frac{(1 + \cos \theta_{\max})^2}{24 \sin^6 \theta_{\max}} \left[6 + 16 \cos \theta_{\max} - 30 \cos^2 \theta_{\max} - 12 \cos^3 \theta_{\max} + 20 \cos^4 \theta_{\max} - 12 \sin^3 \theta_{\max} \cos \theta_{\max} \log \tan (\pi/4 + \theta_{\max}/2) \right] \quad (3-1)$$

where σ_ψ is the standard deviation of the flux distribution across the solar disc.

Equation 3-1 applies to a round absorber tube. The equation for a flat absorber is similar and is also normally distributed in the absorber width. The adjustment in g for radiation at incident angles other than zero is obtained by dividing g by the cosine of the angle of incidence.

3.2 COST MODEL

Trough designs by Honeywell and Sandia are quite similar in concept and both are treated in this section. In adapting the designs of each to large-scale manufacturing techniques the revised Westinghouse designs were nearly the same. Various aspects of both the Honeywell and Sandia troughs, are described in this section. Installed costs for large-scale production are estimated and some parametric relationships (models) are developed.

3.2.1 Honeywell Design

The Honeywell parabolic trough was designed as a test model with the result that certain components are built to closer tolerances for operating efficiencies than for large-scale production and operation. The scale model was built to three-eighths of its intended size. As a result, actual costs of manufacture have no particular relevance as a basis for predicting the costs of large-scale production. The model has been useful, however, by providing insight into ways to design and install large numbers of parabolic collectors.

Some of the characteristics of the Honeywell collector are as follows:

- Rim angle of 115 degrees.
- 1.2 m aperture width by 2.4 m length.
- Portable mount so that the collector could be oriented in either the east-west or north-south directions.
- Alzak surface with an initial specular reflectivity of 81 percent that degraded to 79 percent.
- Various absorbers, one of which is an AMA selective coating, and evacuated area inside a 10 cm glass pipe.
- Sensor and tracking mechanisms that yield accuracies better than 0.1 degree.
- Concentration ratio of 20.
- Operating fluid temperatures up to 300°C.
- Honeycomb sandwich material for the reflective shell, with composite slope errors reducing radiation reaching the absorber by 4 to 5 percent.
- Efficient operation during wind velocities of 13 m/s.

Figure 3-2 shows a drawing of the Honeywell trough as modified by Westinghouse for large-scale production. This drawing served as the basis for the material take-off used in estimating material costs and

Figure 3-2. Westinghouse Adaptation of the Honeywell Parabolic Trough

subsequently equipment, labor and overhead costs. Worksheets used by Westinghouse staff in preparing estimates of these values are contained in Appendix A.

A summary of the estimate for the revised Honeywell trough is shown in Table 3-1. While the Westinghouse worksheets provided the basis for most of the values shown in the table, some modifications and additions were carried out by Colorado State University faculty and Mr. D. A. Watt, consultant to the University. These modifications and additions concerned estimates of manufacturing overhead, base, foundation, motor and gear drives, transportation, and installation costs. In addition, the estimates of absorber costs are obtained from the generalized absorber equations described above.

3.2.2 Sandia Design

The parabolic troughs built by Sandia were intended to explore alternatives for large-scale production that would demonstrate the practicality of their Total Energy concept. As of July 1975, three parabolic troughs had been built and tested, one employing an impregnated plywood shell and another built out of fiberglass. Alzak and back-silvered glass surfaces were attached to these support materials. Much of the cost data available on these troughs was considered by Sandia staff as being higher than would eventually be obtainable, partly through large-scale production and partly through more cost-effective designs.

Some pertinent characteristics of the Sandia trough are:

- East-west orientation to conserve on the cost of field piping.
- Black chrome 4 cm absorber with 1 cm evacuated annulus.

TABLE 3-1

Itemized Costs for the Honeywell Parabolic Trough
with
Modifications by Westinghouse
and
Colorado State University

<u>Item</u>	<u>$\\$/m^2$ (Projected Area)</u> <u>1.22 m Width</u>
<u>Installed Cost, Without Absorber</u>	
Materials	
Shell (honeycomb core)	\$12.40
Reflective surface (back-silvered glass)	3.40
Drive (motor, gear reduction, shafts)	22.60
Sensor	1.80
Labor and overhead to manufacture	12.50
Base and related hardware	3.20
Foundation	19.20
Transportation	1.50
Installation and adjustment	<u>8.00</u>
Subtotal	84.60
Contingency (5 percent)	<u>4.20</u>
TOTAL	\$88.80
<u>(1.9 cm, 90 kg/cm²)</u>	
Materials	
Carbon steel tube	0.50
Borosilicate glass tube	0.30
Flexible couplings	2.40
Other	1.00
Labor and overhead to manufacture and install	<u>2.00</u>
Subtotal	6.20
Contingency (5 percent)	<u>0.30</u>
TOTAL	\$ 6.50
GRAND TOTAL	<u>\$95.30</u>

- Flexible couplings at ends of five 9'x12' panels (note this design may have been subsequently changed).
- 90 degree rim angle.
- Common slope errors of the reflector shell on the order of 0.5 degree.
- Tracking errors less than 0.1 degree.
- Optical sensors with either clock or computer control.
- Measured reflectivities for back-silvered glass of 84 to 88 percent.

Figure 3-3 shows a drawing of the Sandia trough that was prepared by Westinghouse. The principal differences between this drawing and the one for Honeywell are the movable absorber, larger size, and absence of a common drive shaft. Worksheets for estimating the cost of this design are also contained in Appendix A.

A summary of the costs for two sizes of the Sandia troughs is given in Table 3-2. The alternative with the smaller aperture width is made of three 1.22 m x 2.44 m (4' x 8') sections in which the 2.44 m curvature produces an aperture width of 2.13 m and a length of 3.66 m. The larger alternative is made up of three sets of 1.22 m x 4.88 m (14' x 16') sections that yield an aperture width of 4.26 m and a length of 3.66 m. Costs per square meter increase due to the thicker honeycomb core, increased drive costs from the higher torque and greater base costs. Lower foundation costs offset these increases somewhat to the point that the resulting costs are similar. If an absorber of the same dimension is associated with the larger collector, the overall costs would show no meaningful difference. It should be added that the selection of the absorber sizes for the larger-sized reflector maintains the same concentration ratio as for the smaller

Figure 3-3. Westinghouse Adaptation of the Sandia Parabolic Trough

TABLE 3-2

Itemized Costs for the Sandia Parabolic Trough
with
Modifications by Westinghouse
and
Colorado State University

<u>Item</u>	<u>\$/m² (Projected Area)</u>	
	<u>2.13 m width</u>	<u>4.26 m width</u>
<u>Installed Cost, Without Absorber</u>		
Materials		
Shell (honeycomb core)	\$15.90	\$17.50
Reflective surface (back-silvered glass)	3.40	3.40
Drive (motor, gear reduction, shafts)	27.70	30.60
Sensor	1.20	0.60
Labor and overhead to manufacture	11.30	11.40
Base and related hardware	4.80	6.20
Foundation	15.70	10.60
Transportation	1.40	1.50
Installation	<u>8.00</u>	<u>8.00</u>
Subtotal	89.40	89.80
Contingency (5 percent)	<u>4.50</u>	<u>4.50</u>
TOTAL	<u>\$93.90</u>	<u>\$94.30</u>
<u>Absorber</u>	<u>(90 kg/cm²)</u>	<u>(2.5 cm)</u>
Materials		
Carbon steel tube	0.50	1.00
Borosilicate glass tube	0.30	0.60
Flexible couplings	1.60	1.60
Other	0.50	0.60
Labor and overhead to manufacture and installation	<u>1.90</u>	<u>3.00</u>
Subtotal	4.80	6.80
Contingency (5 percent)	<u>0.20</u>	<u>0.30</u>
TOTAL	\$ <u>5.00</u>	<u>\$7.10</u>
GRAND TOTAL	\$ <u>98.90</u>	<u>\$101.40</u>

collector. These sizes are included for illustrative purposes. The optimization program tests alternative collector and absorber sizes to arrive at least cost combinations.

3.2.3 Some Parametric Relationships

Cost-dimensional relationships were estimated for three variables: aperture width, rim angle, and trough length for polar mounts. The data for aperture width were sufficient to establish the general levels of cost for the range of values from 1 to 4 m; but were insufficient to yield a reliable cost curve. But based on general knowledge of the various off-setting cost factors and on previous work of a similar nature in the first STEPS study (3), a probable cost curve was drawn as shown in Figure 3-4. Further analysis and calculations would reveal the shape of the curve more accurately.

Considering the overall costs of the collector units, the effects of changing the rim angle are not great. The major differences are the costs of materials for the shell and reflective surface which vary with the arc length. The values shown in Figure 3-5 should be added algebraically to the cost of those found in Tables 3-1 and 3-2. For example, if the rim angle for the 2.13-m Sandia trough is changed from 90 degrees to 45 degrees, as assumed for the calculations in Table 3-2, $\$1.25/\text{m}^2$ of projected aperture area should be subtracted from total collector costs of $\$93.90/\text{m}^2$. The resulting value would be $\$92.65/\text{m}^2$. If the rim angle is increased to 115 degrees, total collector costs would increase by $\$1.80/\text{m}^2$ to give a new total of $\$95.70/\text{m}^2$.

Should the troughs be mounted in the north-south direction at an angle of 30 degrees (representative latitude for the southwest), the additional costs of earthwork, sketched in Figure 3-6, to mount the

Figure 3-4. Costs of Parabolic Troughs without Absorbers Mounted Horizontally

Figure 3-5. Changes in Costs with Changes in Rim Angle for Various Designs

trough at this inclination would be as shown in Figure 3-7. These estimates of cost are based on unit costs of incremental clearing and grubbing which are additional to that for a horizontal installation, cuts, and fill, select material, compaction, finishing to grade and soil stabilization. The 30 degree angle produces a step-to-rise ratio of 1.73 to 1.0; for the downhill portion a slope of 1.5 to 1.0 was maintained. A roadway is provided at the foot of each slope for ease of installation, servicing and maintenance. Estimates of unit costs for earthwork were obtained from U.S. Bureau of Reclamation staff in Denver, who are experienced in design and construction of large canal systems. The estimate for cut and fill (assuming the fill is adjacent to the cut) is $\$1.05/\text{m}^3$ and that for compaction is $\$0.52/\text{m}^3$. Other sources yielded cost estimates for select material, ($\$5.23/\text{m}^3$), finishing to grade ($\$0.39/\text{m}^2$), land clearing ($\$0.20/\text{m}^2$), and soil stabilization ($1.30/\text{m}^2$).

Figure 3-6. Sketch for Earthwork to Support N-S Collector Orientation

Figure 3-7. Costs of Earthwork for Polar Mounts of 30° for Parabolic Troughs with East-West Spacing Equal to Aperture Width

3.3 OPTIMUM CONCENTRATORS

The minimum costs of concentrators for a large number of design iterations, as characterized by different values of the effective aperture, E , and the image spread measure g , were determined. The effective aperture and image spread measure are the two parameters required to characterize concentrator performance in a STEP System.

3.3.1 Effect of Image Spread

The minimum cost for a parabolic trough as a function of image spread is shown by the solid line in Figure 3-8. A constant value for aperture of 4.15 m in this figure was chosen to show the effect of image spread on concentrator costs. It is observed that the minimum cost for the concentrators is not a strong function of image spread unless a very narrow spread is desired. The accuracy of the surface shape required to achieve a narrow image will increase the production cost significantly. For reasonable surface accuracies, which are readily achieved in the production process, the cost of the concentrator is insensitive to image spread over a wide range. Relaxing the tolerance in surface accuracies has little effect on collector costs.

3.3.2 Effect of Effective Aperture

The change in minimum cost concentrations with effective aperture is illustrated in Figure 3-9 for a fixed E/g ratio of 30. The cost increase for E greater than 4 is due to wind loading. For large aperture widths the tracking drives and structural supports become expensive. A practical limit of 4 to 4.5-m width therefore exists for parabolic troughs.

Figure 3-8. Minimum Costs Parabolic Troughs as a Function of Image Spread

Figure 3-9. Concentrator Cost as a Function of Effective Aperture

3.3.3 Minimum Costs for Concentration

The numbers of minimum cost concentrators resulting from computer runs of concentrator optimization are far greater than the few minimum costs that are illustrated in Figures 3-8 and 3-9. Each of the many minimum-cost concentrators is for a specific set of performance parameters, E and g. Each concentrator characterized by an E and g has an optimum design in terms of aperture width, rim angle, contour and pointing error. Only the values, E and g, are transmitted to subsequent sub-system analysis, and when the optimum E and g pair is identified in an optimum system, the design variables to achieve the particular E and g values can be retraced within the computer program.

3.4 ABSORBER-HEAT EXCHANGER PERFORMANCE MODEL

An evacuated tube with a cylindrical absorber heat-exchanger coated with a selective surface was selected for use in this study for all of the concentrating collectors. The absorptance and transmittance values used are practically achievable. (For the CPC a flat tube and plate absorber was substituted for the cylinder absorber).

The Hottel-Whillier-Bliss equation was changed to be appropriate for analysis of the evacuated tube absorber-heat exchanger. Two forms of the basic model were necessary, one for analysis of sensible heat transfer in pressurized hot water systems and the other for latent heat exchange in steam-generation at the collectors.

For sensible heat addition, the average fluid temperature in the absorber-heat exchanger is given by Equation (3-2).

$$T_{ave} = \frac{I_D EL \alpha G(g, a) (1 - F_R / F')}{U_L A_L} + (T_{in} - T_{amb}) F_R / F' + T_{amb} \quad (3-2)$$

where

- I_D is the direct insolation
 A_p is the aperture diameter
 ρ_{ave} is the average reflectance
 L is the length of the concentrator
 α is the absorptance
 τ is the transmittance of the evacuated tube
 $G(g,a)$ is the normal distribution in the absorber tube diameter, a , with a standard deviation of g .
 U_L is the heat loss coefficient for the absorber
 A_L is the area for losses of the absorber
 F' is the factor that converts the Hottel-Whillier-Bliss analysis equations from average tube temperature to average fluid temperatures.

The factors F' and F_R are given in Equations (3-3) and (3-4) below, where h_o is the heat transfer coefficient. Equation (3-2) is solved by using successive approximations for the fluid properties on T_{ave} .

$$F' = 1 / \left(\frac{1 + U_L}{h_o} \right) = 1 / \left(1 + \frac{U_L}{h_o} \right) \quad (3-3)$$

$$F_R = \frac{\dot{m} C_p (1 - e^{-\frac{-F' U_L A_L}{\dot{m} C_p}})}{U_L A_L} \quad (3-4)$$

The fluid outlet temperature is given by Equation (3-5)

$$T_{out} = \left[\frac{I_D E \alpha \tau G(g, a)}{U_L A_L} + T_{amb} \right] \frac{U_L A_L F_R}{\dot{m} C_p} + T_{in} e^{-\frac{F' U_L A_L}{\dot{m} C_p}} \quad (3-5)$$

and useful heat from the collector is determined from Equation (3-6)

$$Q_u = F' I_D E \alpha \tau G(g, a) - U A (T_{ave} - T_{amb}). \quad (3-6)$$

Equation (3-5) is used to determine the outlet temperature under a given set of design conditions. In the collector performance calculations, the design variables are fixed and the outlet temperature is computed. In the collector optimization calculations, other design variables are fixed and the diameter of the absorber, a , is iterated by using an interval search technique until the desired outlet temperature is achieved.

For latent heat addition, the tube-wall temperature is determined by Equation (3-7).

$$T_w = (D_o h_b T_b + I_D E \alpha \tau G(g, a) + U_L A_L T_a) / (D_o h_b + U_L A_L). \quad (3-7)$$

The equation is solved by successive approximation to determine the correct temperature of the tube wall for the given solar input, climatic condition and fluid flow rate. When the tube wall temperature has been determined, the heat delivered is determined by use of Equation (3-8).

$$Q_u = I_D E \alpha \tau G(g, a) - U_L A_L (T_w - T_{amb}) \quad (3-8)$$

in which

D_o is the outside diameter of the absorber tube, and

h_b is the boiling heat transfer coefficient.

In the collector optimization, the absorber diameter is adjusted with various sets of other fixed design parameters to achieve heat transfer into steam.

3.5 COLLECTOR PERFORMANCE

Because the optimization routine evaluates a large number of different designs, much of the performance information generated by the concentrator and absorber-heat exchanger models is not printed in hard copy form. Detailed information can be examined for selected optimum collectors when desired by using a separate subroutine. The capabilities of the optimization program are described in this section along with a sensitivity analysis of design variables.

3.5.1 Insolation and Ambient Temperature Data

Four typical/days of insolation and ambient temperature data in Albuquerque, N. M. were used in this study. These data are for a typical winter, spring, summer and autumn days. For the purposes of calculating sun position, these days were assumed to occur at the winter and summer solstices, and the spring and fall equinoxes.

The solar insolation and average hourly temperatures of the selected days are shown in Figures 3-10 through 3-13. It will be noted that forenoon insolation levels are not symmetric with the afternoon insolation levels.

3.5.2 Design Example

The design values for a baseline design of a parabolic trough are listed in Table 3-3.

Figure 3-10. Average Hourly Solar Radiation and Ambient Temperatures for December 21 in Albuquerque, N. M.

Figure 3-11. Average Hourly Solar Radiation and Ambient Temperatures for March 21 in Albuquerque, N. M.

Figure 3-12. Average Hourly Solar Radiation and Ambient Temperatures for June 21 in Albuquerque, N. M.

Figure 3-13. Average Hourly Solar Radiation and Ambient Temperatures for September 21 in Albuquerque, N. M.

TABLE 3-3
Design Parameters for Parabolic Trough Example

<u>DESIGN VARIABLE</u>	<u>VALUE</u>
Latitude	33 degrees
Collector type	Parabolic Trough
Orientation and tracking	East-West and North-South Fixed Tilt
Tilt of Collector	Not applicable
Absorber Width	.06 m
Absorptance	.95
Emmittance	.32
Transmittance of covers	.92
Inlet temperature	.40°C
Outlet temperature	300°C
Trough width	2.286 m
Trough length	6.096 m
Rim Angle	65 degrees
Reflectance	.85
Surface Contour Error	.1
Wind Velocity	5 m/s

This example is for a system with pressurized water circulation. In the optimization computerized analysis, both pressurized water and stream circulation systems are considered.

The efficiency of the baseline parabolic trough is illustrated in Figure 3-14 for range of outlet temperatures. The performance illustrated is for noon operation of the collector on September 21. By using noon-time condition and steady-state analysis, cosine, end effects and image spread (spill) at the focus are not reflected in the efficiencies that are shown.

3.5.3 Dynamic Performance

The dynamic effects on collector performance with varying angles of incidence on a parabolic trough which is oriented N-S at a fixed tilt, and delivering fluid at a fixed temperature at 300°C, are illustrated for the 4 selected days of the year in Figures 3-15 through 3-18. In this example the peak power delivered from the collector is less on June 21 and December 21 compared to equinox days because the tilt is set to favor equinox days for energy collection. By changing the tilt angle and temperature of the fluid delivered in the optimization analysis, (as well as other design parameters) eventually an optimum configuration is determined. To determine an optimum system, (i.e. a system which can generate lowest cost energy), a complete analysis of the total system including the transport, storage and power generation subsystem, must be included in the analysis.

The calculated collector efficiencies vary less than 10 percent each day with nearly constant efficiencies from 0900 to 1600 on December 21. The collector performance depicted for the four days are illustrative

Figure 3-14. Efficiency of the Baseline Parabolic Trough

Figure 3-15. Hourly Performance for December 21 for an N-S Parabolic Trough with a Fixed Tilt. Outlet Fluid Temperature 300°C

Figure 3-16. Hourly Performance on March 21 for an N-S Parabolic Trough with a Fixed Tilt. Outlet Fluid Temperature 300°C

Figure 3-17. Hourly Performance on June 21 for an N-S Parabolic Trough with a Fixed Tilt. Outlet Fluid Temperature 300°C

Figure 3-18. Hourly Performance on September 21 for an N-S Parabolic Trough with a Fixed Tilt. Outlet Fluid Temperature 300°C

of dynamic effects on clear sunny days, and for the purpose of this study, which is to compare the performance of four types of linear concentrator systems, four days is satisfactory. Effects of cloudiness are important for system dynamics, to determine an optimized control strategy, but should not be important in comparing linear collector systems because the effects would be similar for the different systems.

Performance curves for the parabolic trough with an east-west alignment are developed on Figures 3-19 through 3-22. The efficiencies of the collector aligned in the E-W direction are significantly less than the collectors on a N-S alignment. Efficiencies vary from about 15 percent in the morning to about 63 percent at noon. Because of the east-west alignment, the peak efficiency is about the same throughout the year and occurs at noon. The reduction in efficiency before and after solar noon is the result of angularity of the solar beam with respect to the surface of the concentrator, which causes image spread at the absorber and end spills. Small effects of reflectance and absorptance with incidence angle would also be reflected in collector efficiency.

At 0750 and 1650 hours the sun's image is more than double that at noon and a larger portion of optical losses are due to spill at the absorber.

A sample of the results of computer calculations for the east-west parabolic trough at the summer solstice with 300°C temperature water is given in Table 3-4. This table contains the same information presented graphically in Figure 3-21 with additional information on the image spread, and mass flow rate through the collector. In the analysis of this section, the mass flow rate is adjusted to obtain the specified

Figure 3-19. Hourly Performance on December 21 for an E-W Parabolic Trough. Outlet Fluid Temperature 300°C

Figure 3-20. Hourly Performance on March 21 for an E-W Parabolic Trough. Outlet Fluid Temperature 300°C

Figure 3-21. Hourly Performance on June 21 for an E-W Parabolic Trough. Outlet Fluid Temperature 300°C

Figure 3-22. Hourly Performance on September 21 for an E-W Parabolic Trough. Outlet Fluid Temperature 300°C

TABLE 3-4

Computations of Performance for a Parabolic Trough on June 21.
 Outlet Temperature is 300°C

Hour	Radiator w/m ²		Effect	Block Effect	End Sp.m.	RHO	g mm	m kg/s	P _o kW	η_c
	DIF	DIR								
5.0	0	0								
5.5	0	439	182	182	124	105	18	.000	0	.000
6.0	0	599	238	238	157	134	19	.001	1	.138
6.5	0	697	289	289	196	167	18	.001	2	.178
7.0	0	773	358	358	257	219	16	.002	2	.231
7.5	0	822	436	436	334	284	14	.003	3	.287
8.0	0	856	520	520	420	357	12	.004	4	.345
8.5	0	879	603	603	509	432	11	.004	5	.403
9.0	0	900	685	685	599	509	10	.005	6	.458
9.5	0	913	757	757	682	580	9	.006	6	.507
10.0	0	924	821	821	759	645	9	.006	7	.549
10.5	0	924	865	865	817	695	8	.007	7	.582
11.0	0	928	901	901	869	739	8	.007	8	.607
11.5	0	928	921	921	905	769	8	.007	8	.622
12.0	0	928	928	928	928	789	8	.007	8	.629
12.5	0	924	917	917	901	766	8	.007	8	.624
13.0	0	924	898	898	865	735	8	.007	8	.609
13.5	0	924	865	865	817	695	8	.007	8	.585
14.0	0	909	808	808	746	634	9	.006	7	.551
14.5	0	889	737	737	664	565	9	.006	6	.509
15.0	0	864	657	657	575	489	10	.005	6	.459
15.5	0	833	571	571	482	410	11	.004	5	.403
16.0	0	796	483	483	390	332	12	.003	4	.345
16.5	0	750	398	398	304	259	14	.003	3	.285
17.0	0	674	312	312	224	191	16	.002	2	.224
17.5	0	568	236	236	160	136	18	.001	1	.160
18.0	0	0								
18.5	0	250	104	104	70	60	18	.000	0	.000
19.0	0	76	35	35						

TOTAL ENERGY IN kJ 226072

outlet fluid temperature from the collector. The increase in the width of the image in the early morning and late evening hours is over twice that at noon.

3.5.4 Sensitivity

The effect on energy output levels from parabolic trough collectors, with N-S and E-W orientations, of changes in some design variables are shown on Figures 3-23 through 3-28.

The sensitivity of collector output energy to absorber diameter variations, from .02 to .12 m, for N-S adjusted tilt collectors are shown for the test days in Figure 3-23. It is evident that energy collected is very sensitive to absorber diameter, if the diameter is small. If the absorber diameter is large, however, the quantity of delivered energy is relatively insensitive to absorber diameter. There is an absorber size that will yield maximum energy output from a parabolic collector, and the slight decrease with larger absorber diameters is due to shading and greater heat loss from a larger absorber surface. Similar sensitivity to absorber diameter is evidenced on Figure 3-24 for a fixed tilt collector, and in Figure 3-25 for a parabolic collector that is oriented with the axis in the E-W direction.

Sensitivity of a parabolic trough collector to rim angle is illustrated in Figure 3-26, with other design parameters fixed. Beyond a rim angle of 45 degrees the energy delivered is not significantly increased. Cost/performance optimizations would therefore seek rim angles closer to 45 degrees than to 85 degrees.

The variations in energy output for the four test days as a function of collector orientation are shown in Figures 3-27 and 3-28 for two

Figure 3-23. Daily Energy Output for a Parabolic Trough with N-S Orientation Adjustable Tilt

Figure 3-24. Daily Energy Output Parabolic Trough with N-S Orientation
Fixed Tilt at Latitude (Albuquerque, N. M.)

Figure 3-25. Daily Energy Output for a Parabolic Trough with E-W Orientation

Figure 3-26. Daily Energy Output for a Parabolic Trough with E-W Orientation

Figure 3-27. Daily Energy Output for Parabolic Trough with Fixed 85° Rim Angle

Figure 3-28. Daily Energy Output for Parabolic Trough with 45° Rim Angle

different collectors with rim angles of 85 degrees and 45 degrees respectively. As previously evidenced the east-west orientation yields significantly less energy from the same collector oriented north-south. The difference between a collector with a fixed N-S tilt and one provided with adjustment requires optimization analysis to determine if the added cost for an adjustable tilt design is cost-effective. For that matter, the cost effectiveness of either N-S designs must be considered against the east-west orientation, given the lower installed cost for the E-W oriented collectors. The optimization results favored the east-west orientation over the north-south orientation on the basis of lowest output energy cost.

3.6 OPTIMUM COLLECTORS

Minimum cost collectors were determined for a variety of different fluid flow rates and outlet temperatures. The optimization procedure requires determination of minimum cost concentrators based on optical analysis and cost, combined with optical and thermal analysis of the absorber-heat exchanger along with cost considerations. There are many minimum cost concentrator designs, virtually one for every combination of effective aperture area, E and image spread factor, g .

3.6.1 Factors Considered

In the optimization analysis, absorber designs were varied along with effective aperture and image spread to determine the minimum cost combination of concentrator and absorber. All of the dynamic effects are included in the optimization with the exception of end spill. With the long collector lengths that result from the optimization, and spills are unimportant with respect to the total energy delivered from the collectors.

For each computerized design iteration of the concentrator, reduction in energy at the focal zone is determined. The reduction in energy is represented by a reduction in effective aperture area. An average increase in image spread was also determined. The two factors, reduced effective aperture and increased image spread, were used first in optimizing the concentrator, and subsequently in optimizing the collector. In optimizing the collectors, an average direct normal insolation was used in the computations in the interest of reducing computation time (and computer costs).

3.6.2 Computation Grids

As in the concentrator optimization, the collector optimization is performed by iteration through "grids" of incremental values for the design variables. The grids are made up of the smallest practical increments commensurate with computer time. The grids are small enough that minimum cost concentrators are satisfactorily identified. A refinement in the computation procedure would be to use "course grids" to first identify the "regions" of optimum collectors and subsequently refine the grid size to "zero-in" on the optimum collector. Unfortunately it was not possible to redefine the variable grids because of the lack of computer funds, but in view of the results of the sensitivity studies, it is unlikely that the optimum collector design was omitted.

3.6.3 Effect of Mass Flow Rate (Power Output)

Several minimum cost and optimum collector designs for various temperatures and mass flow rates are shown on Figure 3-29. For the given mass flow rates and temperatures, an absorber width, effective aperture, image spread and length were selected to produce the required amount of heat to attain the desired temperature with higher mass flow

Figure 3-29. Minimum Cost Parabolic Trough Collectors

rates, the aperture widths of the collectors were very wide and the cost to support the structure against wind loads were high. With higher temperatures there were greater thermal losses, and less power delivered. To maintain low cost collectors, absorber sizes reduced with increasing temperatures, which had its compensating feature of enabling some reduction in aperture size.

3.6.4 Effect of Boiling Temperatures

Boiling temperatures (steam generating absorbers) produce higher average thermal losses from the absorber than pressurized water systems because the temperature of the absorber-heat exchanger is at the boiling temperature whereas in pressurized water systems the average absorber-heat exchanger temperature is approximately the average of the outlet and the inlet temperatures. Thus, the collector efficiencies with steam systems will be lower than for pressurized water systems. To avoid elimination of steam producing collectors from consideration in this study, an emittance of .28 and an absorptance of .97 were selected for the properties of the selective surface. Additionally, the back of the absorber-heat exchanger was insulated to further reduce thermal losses.

Absorbers with selective surfaces, insulation, and evacuation were necessary to produce the high temperatures because of the surface areas in line-focus absorbers are large. Concentration ratios tended to be high for the optimized collectors to compensate for absorber heat losses.

3.7 MINIMUM COST COLLECTOR FIELDS

Once minimum cost collectors were determined for a variety of mass flow rates and fluid temperatures, the collectors were assembled into a collector field within the computer to provide the heat delivery rates

and power production capacity required for cost-effective generation of electricity. The process of assembling the minimum cost collectors into an effective field array consisted of specifying the temperature and mass flow rate through the collectors. The optimization methodology determines pipe sizes in the field and the insulation thicknesses on the pipes. Different optimization runs were made for different fluid temperatures, collector mass flow rates and associated minimum-cost collectors. The results were then compared automatically to select minimum cost collector fields.

In the previous CSU/Westinghouse study it was found that boiling in the collectors was decidedly more cost effective than heating pressurized water in the collector field and using a heat exchanger to generate steam for the turbines. The high heat loss in the return lines with pressurized water systems is a major factor in preference for steam systems. The high temperature in the return lines for pressurized water systems is governed by the pinch point in the heat exchanger.

In the collector fields, a feeder line is used to first add sensible and then latent heat to the fluid in the collector. For line focus collectors, a long collector can be assembled in an E-W alignment with expansion joints. The collector length is traded off with insulation and pipe cost for the expansion joints in the optimization calculations.

The field consisted of rows of east-west oriented feeder lines on either side of a trunk line. The mass flow rate within the trunk line was increased by the mass flow from each pair of feeder/lines on opposite sides of the trunk line. The mass flow in the trunk line thus increases toward the center of the field. The trunk line is duplicated for the other half of the field to yield a rectangular collector field configuration

with the turbine, generator and cooling tower units located in the center of the rectangular array. Separators were used to produce dry steam.

The optimization methodology produced a minimum cost system for a range of heat flow rates and steam temperatures. For each minimum cost field optimum pipe sizes and insulation thicknesses for each feeder line and trunk line was determined. The mass flow rates considered in the study ranged from .08 to 1.28 kilograms per second for each feeder line, and the steam temperatures ranged from 100 to 300°C.

The effective shape of the field was determined by comparing runs with the same temperature but different feeder line. This trade-off showed that feeder lines should be long and trunk lines considerably shorter, principally because the feeder lines were the long collectors. As discussed in the last section, the cost of the parabolic trough collectors increased significantly with increasing collector width. The absorber-heat exchanger costs also increase with absorber width. These factors combine to force narrow collectors and hence lower mass flow rates and longer collectors.

The costs for thermal power from the field at various steam temperatures and plant sizes are shown in Figure 3-30. Thermal power at the lower temperatures is less expensive because of the lower heat losses. As is evident in the figure, there is a large increase in the cost of thermal power when the steam temperature is raised from 250 to 300°C.

An optimum size field to produce steam at lowest cost will not result unless the quantity of thermal power is specified. As the field gets smaller, heat losses will become less. Therefore, if only thermal power is required a very small field would be desirable. However, when

Figure 3-30. Cost Thermal Power from an Optimized Parabolic Trough Collector Field

the turbine-generator subsystem is added a minimum will be evident because larger turbine-generators are cheaper per unit power output.

3.8 MINIMUM COST SYSTEM

The minimum cost per kilowatt hour for three inlet temperatures is shown in Figure 3-31 for plant sizes ranging from 2 to 20 MWe. Minimum storage is assumed and no plant operating and maintenance (O&M) charges are included. O&M costs increase the electrical energy costs by about 1.5 cents per kWh. Minimum cost for electricity generated by a parabolic trough collector field is about 6.5 cents per kWh and with O&M costs, the production price increases to about 8 cents per kWh. The cost of production is not sensitive to plant size, with only 0.5 cents per kWh difference for 2 MWe systems and 20 MWe systems. For parabolic trough collector systems, an optimum configuration can be designed for any plant size in the range studied.

The values of the design variables for the minimum cost parabolic trough STEP system are listed in Table 3-4.

Figure 3-31. Minimum Cost Electrical Energy for Parabolic Trough Collector Systems

TABLE 3-5

Optimum Design for the Minimum Cost
Parabolic Trough Collector System

<u>OPTIMUM STEPS DESIGN VARIABLE</u>	<u>VALUE</u>
Aperture Diameter	2.18 m
Rim Angle	45°
Collector length (feeder line length)	770 m
Surface contour and pointing error	.1°
Absorptance (fixed)	.97
Transmittance (fixed)	.95
Emittance (fixed)	.28
Reflectance (fixed)	.78
Mass Flow Rate of Collectors	.24 kg/sec
Inlet temperature to Collectors	40°C
Outlet temperature (boiling temperature) of collectors	250°C
Wind velocity (fixed)	5 m/sec
Absorber heat exchanger outside diameter	.055 m
Number of collectors (feeder lines) in field	92
Distance between collectors	2.18 m
Mass flow rates out of separators	21.97 kg/sec
Electric Power Plant Size	16.3 MW
Minimum Cost electricity	0.64 \$/kWh
Concentration Ratio	40:1
Average Reflectivity	.78
Turbine Efficiency	.86
Approach to Wet Bulb	3 degrees

4.0 FIXED ABSORBER WITH ROTATING SLATS

Arrays of rotating slats in a plane which is fixed at a slope equal to latitude, like the Sheldahl or ITEK collectors, are evaluated in a solar thermal electric power system in the optimization program. Each slat has a curved reflective surface to reduce image spread at the absorber. Cost and performance models were developed and the concentrating collector was optimized in the STEP system.

4.1 COST MODEL

The ITEK collector is designed to eliminate the problems associated with a movable absorber and high structural costs due to large wind loads on parabolic troughs. A model was built and tested by ITEK to evaluate the technical performance of this type of collector. Production designs, as shown below, would depart in a number of ways from the test model. In the analysis that follows, the standard absorber as described earlier was used in place of the absorber proposed by ITEK to achieve uniformity in comparing different collectors. The cost effectiveness of the ITEK collector in a system can be demonstrated without using the absorber designed by ITEK.

Some relevant aspects of the ITEK collector include:

- A 1.9-cm absorber, without special surface coatings within an insulated 15-cm glass pipe.
- Polar mounting (i.e., module tilted facing south) with slats running in the east-west direction.
- Tracking accuracy within 0.1 degree.
- 5 to 14 mirror slats per module.
- Egg crate polyurethane shell construction to give adequate rigidity for the slats.

- Distance between absorber supports of 6.1 m.
- Seasonal adjustments to improve sun angle.

A drawing of the ITEK collector as modified by Westinghouse for large-scale production is shown in Figure 4-1. The general appearance conforms closely to that contained in an earlier paper by Goralnick (6). In this paper the relationship between the test model and ITEK's proposal for large-scale installation is compared.

Cost summaries for two widths (1.78 m and 3.56 m) of ITEK collectors are given in Table 4-1. The collector is 6.1 m long in both cases with seven 25.4 cm slats in the smaller version and 14 slats of the same size in the larger version. Focal lengths are 1.22 m and 2.44 m and projected aperture areas are 10.84 m^2 and 21.68 m^2 both respectively. Drive and foundation costs have indeed been reduced significantly when compared with the parabolic troughs; but a cost of earthwork is added to allow positioning of the frames at a 30 degree angle facing south. As noted earlier, other possibilities for installing the frames at this and alternative angles were not estimated in this study.

4.1.1 Two Parametric Relationships

Of the various possibilities for testing costs against design changes, only those of aperture and slat widths were carried out. Figure 4-2 shows the drop in costs from $\$92/\text{m}^2$ for an aperture width of 1.78 m to $\$75/\text{m}^2$ for an aperture width of 3.56 m. As before, this parametric relationship is without the absorber. While it is fully recognized that two points are insufficient to trace out a curve, the relationship is a familiar one; and other points would be expected to fall rather closely to the dotted lines.

Section A-A

Note: Effective Slot Width
Totals 1.78m

Figure 4-1. Westinghouse Adaptation of the ITEK Collector

TABLE 4-1

Itemized Costs for ITEK Collector
with
Modifications by Westinghouse
and
Colorado State University

<u>Item</u>	<u>\$/m (Projected Area)</u>	
	<u>1.78 m width</u>	<u>3.56 m width</u>
<u>Installed Cost, Without Absorber</u>		
Materials		
Slats (honeycomb core)	\$ 13.70	\$13.70
Reflective surface (back-silvered glass)	2.60	2.60
Support frame and connections*	5.30	6.80
Drive (motor, gear reduction, shafts)	15.60	9.60
Sensor	1.50	0.80
Labor and overhead to manufacture	9.70	9.60
Base and related hardware	9.00	4.50
Foundation	12.70	6.40
Transportation	1.90	1.80
Installation	9.60	9.60
Earthwork	<u>5.60</u>	<u>6.20</u>
Subtotal	87.20	71.60
Contingency (5 percent)	<u>4.40</u>	<u>3.60</u>
TOTAL	\$91.60	\$75.20
<u>Absorber</u>	<u>(90 kg/cm²)</u>	<u>(2.54 cm)</u>
Materials		
Carbon steel tube	0.60	1.20
Borosilicate glass tube	0.40	0.80
Flexible couplings	1.90	1.80
Other	1.10	0.40
Labor and overhead to manufacture and installation	<u>2.50</u>	<u>4.20</u>
Subtotal	6.50	8.40
Contingency (5 percent)	<u>0.30</u>	<u>0.40</u>
TOTAL	<u>\$ 6.80</u>	<u>\$ 8.80</u>
GRAND TOTAL	<u>\$98.40</u>	<u>\$84.00</u>

* Includes A-frame for absorber

Figure 4-2. Costs of ITEK Collector with Absorber A-Frame but without Absorber

Changes in slat width, or alternatively the number of slats per module, alter unit costs significantly in the extremes. The points shown in Figure 4-3 indicate a band over which changes in costs would vary with changes in slat width. As the slats become smaller in size, the torque becomes less, which reduces drive costs. The cost of the steel skin encasing the honeycomb sections increases because of the greater number of vertical strips. Connecting rods and other costs are not significantly altered over the size range considered in Figure 4-3. The relationship between drive and steel skin costs for the 1.78 m collector is shown below

<u>Item</u>	<u>\$/m² (Projected Area)</u>		
	<u>Slat Width</u>		
	<u>11.9 cm</u>	<u>25.4 cm</u>	<u>59.2 cm</u>
Slats	16.2	13.7	9.4
Drives	16.5	17.1	19.1

As is evident, the principal difference is due to changes in the cost of slats. Were slat widths to become even smaller, the increase in costs would become more dramatic, as can be seen from the slope of the curve in Figure 4-3.

4.2 CONCENTRATOR PERFORMANCE MODELS

The performance model of the rotating slats collector (RSC) includes consideration to design factors and effects that are not required in the parabolic trough model. One effect is the blocking and shading of neighboring slats which reduces the effective collector aperture area.

Calculations of the cosine effect is not as straight forward for this collector as it is for the parabolic trough because the declination

Figure 4-3. General Area Showing Changes in Costs with Changes in Slat Width for ITEK Collectors of 1.78 m and 3.56 m Widths

of the sun is not the only determining factor in slat orientation. The azimuth also affects slat orientation.

The off-normal radiation causes a reduction in the slat aperture. Because each slat has a different orientation with respect to the absorber, the optical analysis is more complex. The cosine effect therefore varies considerably from either a parabolic trough or a flat-plate collector with the same fixed slope.

With an appropriate curvature of the mirror surface for a particular sun and absorber orientation, the same surface curvature causes image broadening for another sun and absorber orientation. The surface curvature for the slats was determined on the basis of maximum total energy collected for the year.

4.2.1 Design Variables and Their Effects

The RSC collector requires low rim angles to minimize image spread and to reduce blocking and shading effects from adjacent slats. The aperture area of the collector is a maximum with the sun normal to the plane of the collector and reduces as the slats rotate. The changes in aperture as a function of the season and time of day are reflected in calculations for effective aperture.

The mirror surface on the slats is assumed to be back-silvered glass and is degraded for dust build-up from a clean surface reflectance of 0.85 to 0.78. Surface contour error is included in the optical model.

The slope of the concentrator was set equal to the latitude, which maximizes energy collection at equinox. A tilt greater than latitude biases energy collection for the winter period and reduces the total energy collected for the year, although a flatter slope may increase slightly the total annual energy collection because of longer periods

of summer sunshine. The possibility of adjusting the slope of the collector periodically was examined but the cost of constructing the collector array with such capability was prohibitively expensive and was not considered further in the analysis.

Slat width has an effect on optical performance because of image spread at the absorber. For off-normal sun angles curvature of the surface also broadens the image. In this study, slat curvature for different collectors was adjusted so that annual energy collection was maximized.

4.2.2 Dynamic Effects

In a dynamic analysis it is necessary to calculate the cosine effect for each individual slat and integrate the effects for all the slats in the collector. The cosine effect for an individual slat is given by Equation (4-1)

$$\cos \theta_r = \frac{\left[\frac{\tan \psi_a \tan \phi_e}{\sin \phi_a} + \frac{h}{r'} \right] \cos \phi_e \sin \phi_a}{2 \tan \psi_a \sin \theta_{se}} \quad (4-1)$$

where:

ϕ_e is solar elevation,

ϕ_a is solar azimuth

ψ_e is absorber elevation,

ψ_a is absorber azimuth,

θ_{se} is slope of the reflector plane in the north-south plane,

h is the elevation of the absorber with respect to the horizontal plant,

r' is the horizontal distance from a point on the absorber to the reflector.

Blocking and shading effects were determined on the basis of slat width, spacing between slats, slope of the collector, and the sun and absorber positions relative to the slat. Calculations for shading and blocking effects were performed for each slat and the total shading and blocking effect for the concentrator was determined by integrating for all the slats and dividing by the number of slats.

The image spread is affected by surface contour and tracking errors, slat width and curvature, distance of the slats from the absorber and the incidence angle with respect to the aperture. Although the slats are curved, the curvature for each slat is assumed to be the same for ease in manufacture. Thus, while the appropriate curvature can be selected for one or two slats, the selected curvature will not be optimum for other slats of the concentrator, and there will be a broadening of the image at the absorber.

A typical ray trace from the rotating mirror surface (RSC) to a fixed absorber is shown in Figure 4-4 together with the parabolic trough collector (PTC). For nonnormal solar incidence angles the reflected path length is longer for the RSC than the PTC, and the reflected image will be broader, therefore, for the RSC collector than the PTC. In the early morning and late afternoon hours, the image of the sun can be twice that for the parabolic trough.

Endspill effects are also taken into account in the dynamic effects. However, the cost-effective collector resulted in long east-west orientation and optical losses at the ends of the concentrator are therefore small.

Figure 4-4. Reflected Path of Non-normal Ray for RSC Concentrator

4.2.3 Analytical Model

Effective aperture, or capability of the concentrator to deliver radiation to the focal zone, is the product of collector aperture, average reflectivity, the cosine factor and the blocking and shading factor. The latter two factors varied significantly depending on the season and time of day.

The distribution of radiation at the absorber is (Gaussian). The equation for the standard deviation of the normal distribution with zero mean is

$$g = \sqrt{\sum_{M=\frac{N}{2}+1}^{N/2} \frac{1}{N} \left[\frac{A_p^2 (4\sigma_\phi^2 + \sigma_\psi^2)}{4 \tan^2 \theta_{\max} \cos^2 \theta_n} + \left\{ R' \cos(-\theta_{se} + \theta_p) - \frac{A_p}{\tan \theta_{\max} \cos \theta_n} \right\}^2 \sigma_Y^2 \right]} \quad (4-2)$$

where N is the number of slats (always even)

A_p is the aperture diameter

$\theta_{\max}$ is the rim angle

σ_ϕ is the standard deviation of the angular surface contour and pointing error distribution

σ_ψ is the standard deviation of the normal distribution approximation to the uniform distribution across the sun's disk

θ_n is the angular deviation of the n th slat from the collector centerline

θ_{se} is the normal elevation of the slat

θ_p is the absorber elevation from the center of the slat

σ_Y is the standard deviation of the normal approximation to the distribution of the angle subtended by the slat as measured from the center of slat curvature. This distribution is considered to be uniform

R' is the radius of curvature of the slat

The effect of each individual slat is summed over the concentrator aperture on a circular absorber tube. An equation for a flat absorber tube was also developed.

The distribution for image spread characterized by Equation 4-2 is for radiation normal the aperture of the collector. The parameter g is then adjusted for other sun positions to account for additional reflected ray path length to obtain the image spread distribution as a function of sun position.

4.3 OPTIMUM CONCENTRATORS

An illustration of how minimum cost concentrators are determined for a wide range of design variables, which are reflected in two parameters, effective aperture and image spread, is given in this section. The effective aperture is the parameter which characterizes the radiation delivered to the focal region and the fraction of reflected radiation intercepted by a given absorber width is determined by g .

4.3.1 Effect of Image Spread

The minimum cost for the RSC as a function of the image spread distribution parameter as shown in Figure 4-5. The minimum cost concentrator is sensitive to the image spread only for small values of g because when close tolerances are required in the production process costs become large. It should be remembered that the minimum concentrators shown on Figure 4-5 are for a fixed effective aperture, E , of

Figure 4-5. Minimum Costs as a Function of the Spread Distribution Parameter for Rotating Flat Concentrators (RSC)

4.15 m, and it is entirely possible that lower cost RSC concentrators exist for different values of E.

4.3.2 Effect of Effective Aperture

The minimum cost RSC concentrators as a function of E (for fixed E/g) is shown in Figure 4-6. For the particular E/g ratio shown, concentrator cost is sensitive to effective aperture E.

4.3.3 Minimum Costs of Concentration

Each point on the minimum cost concentrator curves shown in Figures 4-5 and 4-6 represents a particular concentrator design. Only a few of many hundreds of possible designs are represented by the curves. The minimum cost for a particular E,g pair are conveyed to subsequent optimization of the solar thermal electric power system.

4.4 COLLECTOR PERFORMANCE

The concentrator together with the absorber constitutes a collector, and therefore an absorber-heat exchanger model combined with the optical model of the concentrator is the collector model. The collector performance model is constructed in a manner that enables examination of intermediate results in the process of selecting an optimized collector.

4.4.1 Collector Example

To illustrate the computations that are made within the computerized analysis, it will be useful to discuss the properties of a particular collector. A list of design variables for the collector is presented in Table 4-2. In this example, pressurized water is used as the heat transport medium and the temperature of the water is raised from 40 to 300°C. The flow rate is controlled to achieve a constant temperature

Figure 4-6. Minimum Costs as a Function of the Effective Aperture for RSC Concentrators

TABLE 4-2

Design Parameters for an Example RSC Collector

<u>DESIGN VARIABLE</u>	<u>VALUE</u>
Latitude	33 degrees
Orientation and Tracking	Fixed tilt with movable slats
Tilt	33 degrees
Absorber width	.16 m
Absorptance	.95
Emissittance	.32
Transmittance	.92
Inlet Temperature	40°C
Outlet Temperature	300°C
Concentrator Width	2.286 m
Concentrator Length	6.096 m
Rim Angle	25 degrees
Reflectance	.85
Contour Error	.1 degree
Wind Velocity	5 m/s
Number of slats	20
Radius of Slat Curvature to Focal Length Ratio	2.25

delivering of heat from the collector. The various factors which affect collector efficiency are represented in Figure 4-7, for steady-state conditions with radiation normal to the aperture.

4.4.2 Dynamic Performance

The hourly thermal power delivered from the example collector for four days in different seasons of the year is given in Figures 4-8 through 4-11. Optical and thermal losses for the collector are shown as reductions from the solar power available.

In comparing the thermal power delivered from an RSC collector with the PTC collector, (Figures 3-19 - 3-22) it is evident that there are greater optical losses for the RSC. Blocking and shading in the forenoon as well as the afternoon cause considerable losses. In effect there is a smaller aperture area for the collector as compared to solar noon.

The intermediate results of computations for June 21 for the example collector is shown in Table 4-3. The image spreading in the early morning and late afternoon is very large and the intensity of radiation at the absorber is not sufficient to generate 300°C water at the collector outlet. The performance results could, of course, be different if the temperature is allowed to vary, but for a solar thermal electric power system variations are not desirable for either storage or turbine considerations.

Typical reduction in aperture due to cosine and blocking and shading effects for the RSC are 65 percent, with both larger and smaller factors for various slat width designs. Average reduction factors for the aperture were used to determine the dynamic effects in the concentrator optimization calculations.

Figure 4-7. Steady-state Efficiency for an RSC Collector with Radiation Normal to the Aperature

Figure 4-8. Hourly Performance on December 21 for Rotating Slat Collector Aligned E-W.
Outlet Fluid Temperature 300°C

Figure 4-9. Hourly Performance on March 21 for Rotating Slat Collector Aligned E-W.
Outlet Fluid Temperature 300°C

Figure 4-10. Hourly Performance on June 21 for Rotating Slat Collector Aligned E-W.
Outlet Fluid Temperature 300°C

Figure 4-11. Hourly Performance on September 21 for Rotating Slat Collector Aligned E-W. Outlet Fluid Temperature 300°C

TABLE 4-3

Performance Computations for an RSC Collector on June 21.
Outlet Fluid Temperature, 300°C

Hour	Radiation W/m		Cos. Effect	Block. Effect	End Spill	ρ Effect	g mm	$\dot{m}$ kg/s	P_o kW	η_c
	DIF	DIR								
5.0	0	0								
5.5	0	439	-108	-108						
6.0	0	599	-167	-167						
6.5	0	697	-87	-87						
7.0	0	773	75	75	1	1	599	.000	0	.000
7.5	0	822	267	267	71	60	335	.000	0	.000
8.0	0	856	483	483	210	178	38	.002	2	.188
8.5	0	879	569	569	316	269	34	.003	3	.262
9.0	0	900	654	586	380	323	31	.003	3	.276
9.5	0	913	728	666	483	411	29	.004	4	.333
10.0	0	924	792	735	582	494	27	.004	5	.381
10.5	0	924	837	784	666	566	26	.005	5	.417
11.0	0	928	874	823	742	631	25	.005	6	.445
11.5	0	928	894	845	804	683	25	.005	6	.460
12.0	0	928	901	852	852	724	24	.005	6	.468
12.5	0	924	890	841	800	680	25	.005	6	.462
13.0	0	924	870	820	739	628	25	.005	6	.448
13.5	0	924	837	784	666	566	26	.005	5	.422
14.0	0	909	780	723	572	486	27	.004	5	.385
14.5	0	889	709	649	471	400	29	.004	4	.336
15.0	0	864	628	563	365	310	31	.003	3	.275
15.5	0	833	540	540	300	255	34	.003	3	.263
16.0	0	796	449	449	195	166	38	.002	2	.182
16.5	0	750	244	244	65	55	335	.000	0	.000
17.0	0	674	66	66	1	1	599	.000	0	.000
17.5	0	568	-71	-71						
18.0	0	0								
18.5	0	250	-61	-61						
19.0	0	76	-17	-17						

TOTAL ENERGY IN KJ 136332

4.4.3 Sensitivity

Sensitivity of thermal energy delivered from the collector as a function of the number of slats for a fixed aperture (rim angle) is shown in Figure 4-12. There is little sensitivity for summer days because the sun position is already high before the collector is activated in the morning, so there is less image spreading, and less blocking and shading. In the winter, there is a 5 percent increase in energy delivered from the collector by changing the number of slats from 8 to 50, but only 2 percent for the spring and fall equinoxes.

Sensitivity to absorber diameter is shown in Figure 4-13. Although the absorber diameters are large in comparison to the PTC collector, when the diameter is less than 12 cm, a significant amount of energy is intercepted. Beyond an absorber diameter of 14 to 16 cm, thermal losses from the absorber cause reduction in energy delivered from the collector. It should be recognized that the sensitivity to absorber diameter, while not significant in Figure 4-13, can be different from other minimum cost collector designs.

The variations of output power as a function of rim angle, with the other design parameters held constant, are shown in Figure 4-14. In this illustration, delivery of energy from the collector is sensitive to rim angle only during summer days. Rim angles greater than 36 degrees, with fixed number of slats, will not increase energy delivery because of image spread. If the size of slats is fixed, and rim angle is increased, there will be an increase in energy delivered from the collector; but the rate of increase is not significant as seen in Figure 4-12, and the cost effectiveness of the larger collector is reduced.

Figure 4-12. Sensitivity of the RSC Collector to Number of Slats for Fixed Rim Angle

Figure 4-13. Sensitivity of the RSC Collector to Absorber Diameter, for a Fixed Reflector

Figure 4-14. Energy Delivered from the Collector as a Function of Rim Angle

4.5 OPTIMUM COLLECTORS

Optimum collectors were determined using the performance models and the parametric cost models. For every E,g pair there is a minimum cost concentrator. These minimum cost concentrators are coupled to the absorber-heat exchanger performance and cost to achieve an optimum collector.

4.5.1 Factors Included

Design variables considered in this optimization of collectors are effective aperture, image spread factor, size of absorber and collector length. Concentrator design variables are included in the determination of E and g for minimum cost concentrators. Dynamic effects are also included in the optical and thermal performance computations of the collector.

4.5.2 Minimum Cost RSC Collectors

Mass flow rates ranging from .08 kg/sec up to 1.28 kg/sec combined with temperatures of 100, 150, 200, 250, and 300°C were used in the computations and a minimum cost collector was determined for each mass flow rate and temperature set. Minimum costs of RSC collectors for selected flow rates to deliver 200, 250 and 300°C fluid temperatures are shown in Figure 4-15. The portion of the total cost attributable to the absorber-heat exchanger and concentrator are shown by the heights of the bars. The collector dimensions and the power produced by the collector are given in the sets of numbers above the bars.

It will be noted that as the mass flow rate increases, the collector aperture width also increases because the collector efficiency is relatively insensitive to the higher heat transfer coefficient obtained by

Figure 4-15. Minimum Cost RSC Collectors

the greater mass flow rate. As the collector width increases, the cost of the structure increases significantly because of wind loads, hence large (wide) collectors are not cost effective.

4.5.3 Effect of Fluid Temperature

For the same mass flow rate, the cost per unit thermal power delivered increases with fluid temperature. There are larger heat losses from the absorber at higher temperatures so the power delivered from the collector decreases. If power delivery is to be constant from a given collector, the reflective surface area would have to be larger for the higher temperature collector, and the cost of the structure would increase. In either case, the energy cost rises with temperature.

The cost for thermal power delivered from the RSC collector is shown in Figures 4-16, 4-17 and 4-18 for fixed temperatures of 200, 250 and 300°C respectively and varying mass flow rates. For comparison, bar graphs are also shown for the parabolic trough and movable absorber collectors for the same temperature and mass flow rate range. The power cost increases significantly for a large mass flow rate because the collector and supporting structure must increase to deliver the solar fluid at the absorber to maintain the prescribed fluid temperature.

4.6 MINIMUM COST COLLECTOR FIELDS

The RSC collector field was optimized for two phase fluid flow, with condenser return and steam generation in the absorber. The feeder lines in the collector field are the long absorbers, which are aligned in the east-west direction. Both sensible and latent heat are added at the absorber-heat exchanger to produce 100 percent saturated steam.

Figure 4-16. Minimum Cost Collectors at 200°C

Figure 4-17. Minimum Cost Collectors at 250°C

Figure 4-18. Minimum Cost Collectors at 300°C

Return and delivery mass flow rates and pipe sizes in the field are optimized along with insulation thicknesses. The best arrangement for collectors in the field was for the collectors with the absorber axis aligned in the east-west direction. The power cost delivered from the field for different fluid temperatures at the delivery power is shown in Figure 4-19 for different power plant capacities.

4.7 MINIMUM COST FOR ELECTRICITY

Calculated minimum costs for electricity from a RSC collector field which generates steam in the field is shown in Figure 4-20. Costs do not include operation, maintenance and capitalization. There is not a distinctive difference in electricity costs either as a function of plant capacity or steam temperature. However, electricity production cost is lowest for steam plants at 250°C, and plant sizes of above 20 MW_e appears most favorable for a STEPS plant with RSC collectors. Electricity cost increases slightly for smaller plant capacities largely because of costs for smaller turbine generator-cooling tower subsystems.

A comparison of minimum electricity costs from an RSC solar power plant and PTC plant is shown in Figure 4-21. Both systems are for steam plants at 250°C, and clearly the PTC system has the potential to produce electricity at a lower cost than an RSC system. The optimum plant size is about 20 MW_e for both systems.

Figure 4-19. Minimum Cost for Power Produced from RSC Collectors in an Optimized Field

Figure 4-20. Minimum Cost Electricity from an RSC STEPS Plant

Figure 4-21. Electrical Energy Costs for Optimum RSC and PTC Power Plants

5.0 MOVABLE ABSORBER CONCENTRATOR (MAC) SYSTEM

Russell et al. (9) describe the General Atomic concentrator concept in the following way: "A nonparabolic mirror that has no off-axis aberration has been discovered.... This unique surface focuses parallel incident light into a sharp line focus, regardless of the incident light direction. The locus of this line focus is a cylinder.... The mirror is composed of long, narrow flat reflecting slats arranged on a reference cylinder. The angles of the slats are fixed so that the focal distance of the array is $2R$, where R is the cylinder radius." It can be shown that (1) in the limit of zero-width slats the focus is sharp for parallel light of any incident direction, and (2) the line focus always falls on the reference cylinder, so that a heat collector can be a pipe supported on arms that pivot at the center of the reference cylinder. It is this concept that makes possible a Fixed-Mirror Solar Concentrator (FMSC)."

The conceptual advantage of such a collector is that concentrations of solar energy can take place without having to move the reflector. Instead, only the much smaller absorber pipe must rotate in response to the movement of the sun. General Atomic has built a test model of the FMSC using thin slats mounted in a structural frame. To improve upon the costs of this model, General Atomic staff carried out preliminary work on the possibility of placing the reflective slats in an earth trench and moving the linear absorber by long rotating arms. The intention is to make use of low-cost, large-volume earth moving and paving equipment. A cross-sectional sketch is shown in Figure 5-1.

Design and construction possibilities, obtained from General Atomic staff and reports, include the following:

Figure S-1. Cross Section of General Atomic Concentrator for $\theta = 120^\circ$

- With the trough running east and west, the controlling axis (for determining slit placement) would be tilted 15 degrees from vertical.
- Aperture width on the order of 8 to 10 m.
- Arc angles for the extremities of the trough from 90 degrees to 120 degrees.
- Stepped reflectors set in concrete, soil cement, or asphalt.
- Reflectors of aluminized plexiglass with reflectivities of 85 percent.
- Trough lengths on the order of 1,000 m.
- Laser guided controls to obtain longitudinal slope accuracies of 1 in 360.
- Paving and earth moving equipment to be rapid, large-scale, and automated.
- Concentration ratios in the range of 50.

5.1 COST ESTIMATION PROBLEMS

Construction design and cost estimating tasks for this collector were divided between Westinghouse and CSU staffs. Westinghouse was responsible for the rotating absorber, absorber mounts, and reflective material. CSU investigated the construction of the trough with technical and costing assistance from the Plan Coordination and Reports Department in the United States Bureau of Reclamation in Denver.

Establishing the basis for estimating the trough costs proved difficult. The difficulty arose in attempting to decide how the cross sections, which require step construction, could be built in an accurate yet rapid manner. Earth shapers of the type commonly in use in large irrigation canal projects apparently are not able to meet cross-sectional tolerances much greater than 10 cm even though tolerances of 1 in 360 can on average be obtained in the longitudinal direction. Another problem is in finding a continuous concrete laying process (e.g., rail

guided, slip forming as in highway pavement construction) that would be able to shape the steps. Staff from the Bureau of Reclamation were skeptical of the possibility. Precast sections poured at the site and lowered in place would be a possibility if slip forming would not work. Settlement and heaving are not considered to be a serious problem, although some differential vertical movement might take place at construction and contraction joints.

To be able to proceed with the optimization process, assumptions were made that (1) cross-sectional accuracies of 2.5 cm can be met, (2) longitudinal accuracy will be 1 in 360, and (3) nonreinforced concrete of minimum 7.6 cm thickness can be poured using slip-forming methods.

5.2 PARAMETRIC RELATIONSHIPS

Cost summaries of two versions of the MAC collectors are presented in Table 5-1. Common characteristics of these two types include θ angle of 90 degrees, slot width of 40 cm and collector length of 1,000 m, reflector radii are 9.15 m and 18.30 m. Under the assumptions used in estimating construction costs, the collector with the smaller radius has overall costs similar to a large RSC Collector. As the radius of a MAC Collector increases, absorber costs become substantially higher while reflector costs (including absorber drives and supports) remain about the same.

Cost-dimensional relationships were estimated for three variables: radius, rim angle, and step width. The nature of changes in costs for different radii can be seen from the results tabulated in Table 5-1. Costs changes for different rim angles are not significant. A reduction in $\theta_{\max}$ from 45 degrees to 35 degrees causes a reduction of only $\$1/\text{m}^2$ of projected area and an increase from 45 degrees to 60 degrees causes unit

Table 5-1. Itemized Cost for MAC Collector

(θ=90°, Slat width = 40 cm)

	Radius	
	9.15 m	18.3 m
	\$/m ² (Projected Area)	
<u>Reflector Trough</u>		
Earthwork	\$ 4.60	\$ 6.50
Concrete step	17.40	17.40
Reflective surface	4.00	4.00
Drainage	5.00	5.00
Miscellaneous	1.00	1.00
Contractor's overhead (20 percent)	6.40	6.80
Subtotal	38.40	40.70
Contingency (10 percent)	3.80	4.10
TOTAL	<u>\$42.20</u>	<u>\$44.80</u>
<u>Materials</u>		
Support for absorber	6.80	8.00
Drive for absorber	8.90	6.30
Transportation	1.80	2.70
Labor and overhead to manufacture	7.80	7.70
Installation	3.00	3.00
Subtotal	28.30	27.70
Contingency (5 percent)	1.40	1.40
TOTAL	<u>\$29.70</u>	<u>\$29.10</u>
	(901g/cm ²)	(25.4 cm)
<u>Absorber</u>		
Glass pipe	5.90	17.80
Carbon steel tube	1.60	5.00
Flange, elbows, bellows, seals	2.30	5.30
Miscellaneous hardware	0.10	0.20
Labor and overhead to manufacture and install	2.50	4.50
Subtotal	12.40	32.80
Contingency (5 percent)	0.60	1.60
TOTAL	<u>\$13.00</u>	<u>\$34.40</u>
GRAND TOTAL	<u>\$84.90</u>	<u>\$108.30</u>

Figure 5-2. General Atomic Collector Changes in Cost with Changes in Rim Angles for Two Radii (R)

Figure S-3. General Atomic Collector with Percent Contour Changes in Cost with Changes in Step Lengths for Two Rim Angles (θ)

costs to increase by less than \$3.00 (see Figure 5-2). The largest change in unit costs arises by varying step widths as shown in Figure 5-3. This response is the result of the effect of step width on the amount of concrete to be poured.

5.3 CONCENTRATOR PERFORMANCE MODELS

The performance models for the MAC collector include optical and thermal effects. Blocking is significant at some sun positions, and as the focal point moves the average distance from a particular reflector surface to the focal point changes. The effects of changing sun position with respect to the fixed mirrors are also considered in the analysis.

5.3.1 Design Variables and Their Effects

The rim angle of the collector affects the total path length of a ray from the mirror to the absorber. The rim angle is also interrelated with blocking and shading effects of adjacent slats. Aperture diameters considered are much larger for the MAC collectors than other collectors in this study because of construction methods.

Wind was considered for the structural support members of the absorber-heat exchanger. Thus, absorber diameter, which is determined by reflection dimensions, bears importantly on the cost of the collectors. To facilitate the analysis, additional cost for the absorber supports was considered as part of the concentrator costs.

The reflective material on this collector was assumed to be back silvered glass as for the other concentrators. Surface contour and pointing errors are considered in the model and also slat width. Slat width has a significant effect on image spread because the slats are flat and dominated optimized concentrator costs.

The collector aperture was tilted to be nearly normal to the sun. However, the slope was not as large as that required for the RSC collector, primarily because there was less blocking and shading if step heights could be kept small.

The term tangent slat angle is used to indicate the position of the slat that is tangent to the circle described by the circular collector segment. By specifying this angle, the position of the focus on the radius of the circle is located. From the tangent slat angle the orientation of the other slats positioned around the circular segment can be established. The location of the tangent slat angle on the circumference also influenced blocking and shading effects of the concentrator.

Slat width has a significant effect on the spread of the image at the focus. Due to construction techniques assumed, and cost functions that were developed, a wide range of slat width was considered. The formed concrete construction method for fabricating the concentrator favored small slat widths in terms of cost.

5.3.2 Dynamic Effects

The position of the sun, or more precisely the sun elevation angle, together with the collector slope and tangent slat angle determine the location of the focus on the radius of the circle. The dynamic effects include cosine and cosine angles, blocking and shading factors. Endspill is modeled using the calculated distance from the slats to the absorber averaged over the number of collector slats, although the effect is insignificant for the long lengths of collectors which resulted from the optimization analysis.

Because the focus is located on the circumference of a circle which circumscribes the collector, the distance of a reflected ray to the absorber is considerably longer than for either the PTC or RSC Collector. Thus, image spread is significantly affected by sun elevation and azimuth. A pictorial representation of the path length of the reflected ray is shown on Figure S-1 in comparison to the PTC and RSC Collectors.

5.3.3 Analytical Models

The effective aperture for the MAC Collector is the product of the aperture, average reflectivity, cosine factor, and the blocking and shading factors. Effective aperture was therefore a function of sun position. The image is normally distributed at the focus. The equation for the standard deviation, g , for each slat in the concentrators is:

$$g_n = A_p \sqrt{\frac{\left[\frac{1 + \cos(\beta + \theta_n)}{2} \right] \left[4\sigma_\phi^2 + \sigma_\psi^2 \right]}{\sin^2 \theta_{\max}}} \quad (5-1)$$

where β is the angular position of the absorber support with respect to the vertical. The angle β is equal to twice the solar zenith minus three times the sum of the collector slope plus tangent slat angle.

If the slat width is sufficiently small, the following approximation is used:

$$g = A_p \sqrt{\frac{4\sigma_\phi^2 + \sigma_\psi^2}{2 \sin^2 \theta_{\max}}} \left[1 + \cos \beta \cot \theta_{\max} \ln \tan \left(\frac{\pi}{4} + \frac{\theta_{\max}}{2} \right) \right] \quad (5-2)$$

In equation (5-2) slat width is assumed to have no effect. If slat width has an effect on image spread, Equation (5-1) must be summed over all the slats in the concentrator to determine the image spread at the target.

Equations (5-1) and (5-2) apply to a circular absorber tube. Equations appropriate for a flat absorber tube were also developed but not used in the optimization because the absorber was limited to circular pipes.

5.4 OPTIMUM COLLECTORS

5.4.1 Effect of Image Spread

The effect of image spread on minimum costs for a fixed effective aperture of 4.15 meters is shown on Figure 5-5. The comparative costs for these types of collectors are only illustrative for a fixed-effective aperture. The relative costs for other effective operations in the analysis change, and differences may not be as large as indicated on Figure 5-4.

Figure S-4. A Reflected Ray for MAC Collector in Comparison to PTC and RSC Collectors

Figure 5-5. Minimum Cost as a Function of Image Spread

Because of the nature of construction, as previously explained in the section on costs, the MAC Collector shows insensitivity to increase in image spread. The value of g which yield the smallest cost per unit aperture width is about 0.14 m.

5.4.2 Effect of Effective Aperture

The minimum cost of MAC Concentrators as a function of effective aperture, for a fixed E/g ratio of 30, is shown in Figure 5-6. Because the MAC Collector is not significantly affected by wind loading, costs increase moderately with aperture width. Wind loading affects only the moveable absorber support structure (which was considered part of the concentrator for purpose of simplifying the analysis).

5.5 COLLECTOR PERFORMANCE

Some intermediate results of the optimization process using collector performance models are discussed in this section. In the optimization analysis a considerable number of designs are iterated and it is not practical to display intermediate results for more than a few collectors. It should be remembered that the results are typical, and are not specifically for the optimized unit for the entire solar plant.

5.5.1 Design Example

The collector selected to illustrate optical and thermal performance is described in Table 5-2. A pressurized water system is used for the heat transfer and transport fluid and the mass flow rate is adjusted during the day to achieve a constant outlet fluid temperature from the collector. Optical and thermal losses for the collector are illustrated in Figure 5-7. Losses include cosine, slat blocking and shading, endspill, reflectance, transmittance absorptance, spill at

Table 5-2. Design Parameters for MAC Collector

<u>DESIGN VARIABLE</u>	<u>VALUE</u>
Latitude	33 degrees
Orientation and Tracking	East-West with tracking absorber
Aperture Slope	33 degrees
Tangent slat angle	0 degrees
Absorber width	.20 m
Absorptance	.95
Emittance	.32
Transmittance	.92
Inlet fluid temperature	40 degrees
Outlet fluid temperature	300° C
Concentrator width	2.286 m
Concentrator length	6.096 m
Rim angle	55 degrees
Reflectance	.85
Contour error	.1
Wind velocity	5 m/s
Number of slats	301

the focus and thermal losses. The illustration on Figure 5-6 is for steady state conditions at solar noon on September 21 and March 21.

5.5.2 Dynamic Performance

Thermal power delivery from the example MAC Collector at 300°C for winter and summer solstices and spring and fall equinoxes are shown on Figures 5-8 through 5-11. The collectors are oriented E-W and the aperture is tilted toward the south. The efficiency of the MAC Collector is considerably less than for the PTC and RSC Collectors. Spread losses are the largest factors which affect efficiency. A table of half-hourly values for the day, June 21, is presented in Table 5-3.

5.5.3 Sensitivity

The effect of absorber widths from 10 to 30 cm on thermal energy delivery rate from the collector is depicted on Figure 5-12. There is an optimum absorber diameter for winter, spring and fall, but in summer and winter a 30 cm diameter absorber produces the largest rate of heat delivery. For the example collectors, the aperture is sloped at latitude and the effective aperture is largest on equinox days. When the absorber is too small, radiation is not intercepted, hence there is reduction in energy delivered from the collector. Beyond 20 cm there is little sensitivity to absorber diameter size.

There is greater sensitivity of the MAC Collector to tangent slat angle. Variations in thermal output for 3 aperture slope angles are shown on Figures 5-13 and 5-14. There will be a different optimum tangent slat angle for each aperture tilt angle and depending upon the time of year, 50 to 100 percent change in energy output is possible for small design changes in tangent slat angle.

Figure 5-6. Concentration Cost as a Function of Effective Aperture

Figure 5-7. Optical and Thermal losses for MAC Collector for Steady State Conditions at Solar Noon on March 21 and Sept. 21

Figure 5-8. Hourly Performance for MAC Collector on December 21
Outlet Fluid Temperature 300°C

Figure S-9. Hourly Performance for MAC Collector on March 21
Outlet Fluid Temperature 300°C

Figure 5-10. Hourly Performance for MAC Collector on June 21
Outlet Fluid Temperature 300°C

Figure S-11. Hourly Performance for MAC Collector on September 21
Outlet Fluid Temperature 300°C

Figure 5-12. Sensitivity of Collector to Absorber Width for a MAC Collector (Tilt, 30°), Tangent Slat Angle, 0°)

Figure 5-13. Sensitivity of MAC Collector to Tangent Slot Angle for Collector Tilt of Zero Degrees

Figure 5-14. Sensitivity of MAC Collector to Tangent Slant Angle for Collector Tilt of 30 Degrees

Table 5-3. Performance Computations for the MAC Collector on June 21. Outlet Fluid Temperature, 300°C

Hour	Radiation (W/m^2)		Cos	Blocking &	End	P	g	n	P _o	η_c
	Dif	Dir	Effect	Shading	Spill		mm	kg/s	kW	
5.0	0	0								
5.5	0	439	-53	0						
6.0	0	599	-0	0						
6.5	0	697	83	0						
7.0	0	773	184	152	110	93	240	.000	0	.000
7.5	0	822	289	242	157	133	118	.000	0	.000
8.0	0	856	393	295	193	164	81	.000	0	.000
8.5	0	879	491	386	265	225	65	.000	0	.000
9.0	0	900	584	482	353	300	55	.000	0	.000
9.5	0	913	665	565	440	374	50	.002	2	.178
10.0	0	924	734	634	524	445	46	.003	3	.241
10.5	0	924	783	685	596	507	44	.003	4	.283
11.0	0	928	822	724	663	563	42	.004	4	.315
11.5	0	928	844	747	715	608	42	.004	4	.331
12.0	0	928	851	754	754	641	41	.004	4	.340
12.5	0	924	841	744	712	605	42	.004	4	.334
13.0	0	924	819	721	660	561	42	.004	4	.318
13.5	0	924	783	685	596	507	44	.003	4	.290
14.0	0	909	722	624	515	438	46	.003	3	.246
14.5	0	889	647	550	429	364	50	.002	2	.181
15.0	0	864	561	462	339	288	55	.000	0	.000
15.5	0	833	465	365	251	214	65	.000	0	.000
16.0	0	796	365	274	179	152	81	.000	0	.000
16.5	0	750	263	221	143	122	118	.000	0	.000
17.0	0	674	160	133	96	81	240	.000	0	.000
17.5	0	568	68	0						
18.0*	0	0								
18.5	0	250	-30	0						
19.0	0	76	-18	0						

TOTAL ENERGY IN KJ 70692

There is considerable loss in optical efficiency for a Mac collector from blocking and shading of adjacent reflector surfaces for an E-W alignment. Averaged over a day, as much as 50 percent reduction in effective reflector surface can occur, and for a fixed geometry, there may be as much as 50 percent difference between seasons of the year.

5.6 OPTIMUM COLLECTORS

The optimum collectors were determined using the sequential optimization methodology, the performance models illustrated in the previous section and the parametric cost models. Dynamic effects were included in the optimization process, although end spill was neglected because of the very long (1000 m) collector lengths.

5.6.1 Computer Methodology

Design variables were iterated over a "grid" of values to optimize the concentrator for each E and g pair of values. Using a procedure similar to the PTC and RSC collectors, the minimum cost design was then identified and these concentrations were coupled to absorbers and absorber variables were iterated to derive an optimum collector.

5.6.2 Effect of Mass Flow Rate

Sample MAC collectors that produce thermal energy at minimum cost are shown on Figure 5-15. Costs range from \$380/kW_t to \$500/kW_t, with lower costs for smaller concentrators, absorbers and flow rates. The costs increase with increasing temperature as expected. Steam systems are better choices for collectors than pressurized water systems.

5.6.3 Comparisons with PTC and RSC Collectors

Maximum cost MAC collectors are compared with parabolic trough and rotating slat collectors for three fluid temperatures in Figures 5-16, 5-17 and 5-18. At all three temperatures the MAC collectors

Figure 5-15. MAC Collectors that Produce Thermal Power at Minimum Cost

Figure 5-16. Comparisons of Collectors for Minimum Cost Thermal Power Delivery at 200°C

Figure 5-17. Comparison of Collectors at 250°C

compare favorably with the RSC collectors, and depending upon flow rate, the costs for thermal power delivered from the collectors are higher than the PTC collectors. For large collectors and large mass flow rates through individual collectors, the MAC collector is more economical because there is less sensitivity of cost to aperture width compared to the PTC or RSC collectors. At 300°C fluid delivery temperature, large RSC and MAC collectors are not economical compared to the parabolic trough, and are not shown on Figure 5-18.

5.7 MINIMUM COST COLLECTOR FIELDS

Collector field dimensions and pipe layouts were optimized on the basis of cost for thermal energy delivered to the power station. Because MAC collectors were insensitive in cost to mass flow rates, hence to aperture width and collector length, long collectors with high mass flow rates were found to be advantageous in the field layout. For all linear collectors steam generation at the collector was more economical than circulating pressurized water in the field with steam generation at a localized heat exchanger at the turbine.

Curves for cost of thermal power delivered to the turbine or storage are shown in Figure 5-19. Except for changes in temperature, there is no sensitivity in cost for thermal power delivery with plant capacity. Thermal energy produced in the field at 250°C is about 13 percent cheaper than at 200°C and 17 percent lower in cost than at 300°C. The cost for storage and the turbine-generator-cooling tower operation at the respective temperatures will determine the preferred operating temperature and plant size.

Figure 5-18. Minimum Cost of Collectors at 300°C

Figure 5-19. Costs for Thermal Power Delivered to a Central Station from Optimized MAC Collectors and Field Arrangements

5.8 MINIMUM COST ELECTRICAL ENERGY

The minimum costs for electrical energy produced from MAC power plants for three temperatures are shown in Figure 5-20. The optimum operating temperature is 250°C, although there is little significant difference compared to power plants operating at 200 and 300°C. For the plant capacities considered in this study, the larger power plants produce lower cost electrical energy, principally because long collectors can be used in the field.

Electrical energy costs for three types of power plants using PTC, RSC and MAC collectors are shown in Figure 5-21. Parabolic trough collector systems are clearly more economical than RSC or MAC power plants. Generally, electrical energy production costs are insensitive to plant capacity, although for the plant capacities of 2 to 3 MW_e energy cost is about 10 percent greater than for power plants of 20 to 30 MW_e size.

Figure 5-20. Minimum Costs for Electrical Energy from MAC Power Plants

Figure 5-21. Comparison of PTC, RSC and MAC Power Plants

6.0 COMPOUND PARABOLIC CONCENTRATOR (CPC) SYSTEM

The CPC concept comes in many versions. The one considered in this study was evaluated by Arthur D. Little (7) and Chamberlain Manufacturing Corporation (8). The technical advantage of the CPC collector is the ability to concentrate solar energy through single and multiple reflections onto a narrow rectangular area. This is accomplished without moving either the reflectors or absorbers. Concentration ratios of 10 or less are considered practical although at 10, the concentration ratio is lower than those found optimal for other line-focus concentrating collectors.

To generate the high temperatures desired for solar thermal electric power systems, and to be consistent in the performance and cost analyses with the other collectors in this study, only an evacuated tube absorber heat exchanger was considered. Instead of a cylindrical absorber within the evacuated tube a flat absorber was assumed. Because it was not possible to generate sufficiently high temperature fluids from the CPC collector to compare with the other linear collectors, analysis of the others were extended to temperatures appropriate for the CPC. The parabolic trough produced lowest cost thermal and electrical energy at temperatures below 200°C so comparisons in this section are made only with the PTC system.

6.1 COST AND COST MODEL

A cost analysis carried out by Chamberlain produced a target selling price for uninstalled collector units in the range of \$97 to \$150 per square meter. An assumed production rate of 460,000 m²/yr is about one-quarter of that assumed for this study. With economies in the scale of

production, a somewhat lower price for the units would be expected. Differences in cost estimates between Chamberlain and this study are not significant when foundation and installation costs are added to the Chamberlain estimates.

A review of the A.D. Little and Chamberlain reports provided the following information about the CPC collector:

- South-facing mount with the parabolic troughs running east and west.
- Seasonal adjustments range from daily to twice a year to maintain a minimum 7 hours of collection time per day, depending on the concentration ratios.
- Overall box dimensions of 1.22 m x 2.13 m and 30 cm thick.
- Optical concentration ratio of 5.3.
- Radiation acceptance half angle of 9 degrees.
- Entrance aperture of 11 cm.
- Exit aperture of 2 cm.
- Height of parabolic reflectors of 17 cm.
- Length of parabolic reflectors of 1.83 m.
- Serpentine copper tubing with the top surface coated with black chrome.
- Boxes containing the CPC units covered with glass, insulated, but not evacuated.
- Reflectors roll formed from coiled reflective grade aluminum sheet (Aizak).

The exploded diagram contained in the Chamberlain report served as the basis for the design modifications and material take-offs carried out by Westinghouse. The Chamberlain diagram is shown in Figure 6-1 and the details on material selection and manufacturing equipment are provided in the Appendix. The Westinghouse design involves a projected area of 1.39 m^2 from net dimensions of 0.91 m x 1.52 m. The absorber

Figure 6-1. Concentrating Flat Panel (CFP) Design for Mass Production.
(Source Chamberlain Manufacturing Corporation)

is 2.54 cm steel tubing encased in the collector box which is covered by double strength float glass. The units are mounted in groups of 16 on fixed steel supports inclined at an angle of 30 degrees.

The summarized cost estimates for CPC collectors of 0.91 m x 1.52 m (1.39 m^2) and 1.22 m x 2.44 m (2.97 m^2) are given in Table 6-1. The absorber which is an integral part of the collector module has been separated in detailing the costs for convenience of this study. Labor and overhead costs are higher for CPC modules than for the other concentrators (20 percent versus 15 percent) because of the more complex nature of the design. The modules are assumed to be mounted on 3.66 m x 7.32 m steel frames. Part of the framing would be attached to the CPC boxes and the remainder would have to be assembled in the field.

Of the different possibilities for testing cost against design changes, only those of aperture widths and collector heights were carried out. Figure 6-2 shows a drop in cost of the concentrator from $\$65/\text{m}^2$ for an aperture width of 5.7 cm to $\$58/\text{m}^2$ for an aperture width of 7.6 cm. These costs do not include the absorber cost. Cost per area decreases with increasing aperture width because of reduced housing, reflector, and structural supports. While it is fully recognized that two points are insufficient to trace out a curve, other CPC designs are projected to be close to the dotted lines from experience in costing other concentrators in this study. Unit cost variation as a function of reflector depth (height) is shown on Figure 6-3.

Figure 6-2. Costs of CPC Collector without Absorber for Varying Aperture Widths
(Constant aperture area = 1.39m²)

Figure 6-3. Changes in Cost with Changes in Height of the Collector.

Table 6-1. Itemized Costs for the CPC Collector with Modifications by Westinghouse and Colorado State University

<u>Item</u>	<u>\$/m² (Projected Area)</u>	
<u>Installed Cost Without Absorber</u>		
	<u>(0.91 m x 1.52 m)</u>	<u>(1.22 m x 2.44 m)</u>
Materials		
Collector housing	\$14.60	\$12.00
Reflector	11.90	10.50
Glass cover	4.20	4.30
Structural supports	6.80	4.10
Labor and overhead to manufacture	7.80	7.80
Base and related hardware	1.00	1.00
Foundation	8.60	8.60
Transportation	2.40	2.00
Installation	<u>5.00</u>	<u>4.50</u>
Subtotal	62.30	54.80
Contingency (5 percent)	<u>3.10</u>	<u>2.70</u>
TOTAL	<u>\$65.40</u>	<u>\$57.50</u>
<u>Absorber</u>	<u>(2.54 cm)</u>	<u>(2.54 cm)</u>
Materials		
Carbon steel tube	7.20	6.80
Other	2.30	2.10
Labor and overhead to manufacture and connect	<u>5.00</u>	<u>3.50</u>
Subtotal	14.50	12.40
Contingency (5 percent)	<u>0.70</u>	<u>0.60</u>
TOTAL	<u>\$15.20</u>	<u>\$13.00</u>
GRAND TOTAL	<u>\$80.60</u>	<u>\$70.50</u>

6.2 PERFORMANCE MODELS

The analytical equations for the compound parabolic concentrator used in this study were adapted from the equations developed by the Argonne National Laboratories in published reports. In contrast to the other concentrators examined in this study, existing analytical work was in a form appropriate for the methodology used in this study. However, some of the analytical results are approximations, particularly in regard to the division of direct and diffuse radiation, and the effects of truncation.

Modifications to the methodology of analysis were required because the CPC is not an imaging concentrator. In particular, the image spread parameter, g , developed for the other concentrators, has no relevance to the CPC. There are radiation distribution effects over the absorber surface but there is no "spill" around the absorber. To accommodate these differences, the fraction of radiation intercepted by the target of a CPC was fixed at unity. The width of the absorber was used as a parameter to characterize CPC performance in place of g . The absorber (target) width together with the effective aperture is the parameter pair that characterizes the performance of the concentrator.

6.2.1 Design Variables and Their Effects

The acceptance angle for the CPC is substituted for rim angle. For a full CPC, the acceptance angle governs the amount of solar radiation that is intercepted. All diffuse as well as direct radiation within the acceptance angle reaches the absorber. Radiation outside the acceptance angle is not concentrated by a full CPC. For a truncated CPC, some radiation outside the acceptance angle reaches the absorber, however it

is neglected in the study because the effect is small on the total radiation at the target.

The aperture width of the CPC is consistent in definition with the previous concentrators. Surface contour and pointing errors are not considered in the analysis of the CPC because the CPC is fixed and surface contour errors of the parabolic sections had minor effect on the reduction of radiation which reaches the absorber.

The determination of average reflectance for the CPC is more complex than for other concentrators. Total radiation on the absorber consists of radiation reflected from the parabolic segments of the CPC as well as radiation directly on the absorber. Analytical equations developed by Argonne National Laboratories for characterizing the average or effective reflectance for various CPC designs were used in this study. The number of compound parabolic concentrators in the collector container was varied.

6.2.2 Dynamic Effects

The time of day and time of year significantly affect the thermal performance of the CPC. Since the CPC is at a fixed slope, the cosine effect is the same as for a flat-plate collector with the same orientation. Blocking and shading occur when the solar beam is outside the acceptance angle and endspill occurs when the CPC box is narrow. Endspill is not significant, however, because of the relatively shallow depths of the concentrators.

6.2.3 Analytical Model

The expression for g for the CPC is straightforward. Because there is no spreading of the image, the size of the absorber is the width of the radiation on the absorber. The effective aperture is the product of the

aperture of the concentrator, average reflectance and the cosine and blocking effects. The average reflectivity of the concentrator is a function of the geometry of the concentrator and the cosine effect is a function of sun position relative to the fixed concentrator.

6.3 OPTIMUM CONCENTRATORS

Minimum cost concentrators were identified for effective aperture and absorber width pairs, with the numbers of CPC reflectors in a collector unit as a third variable. Iterations of design variables were made in the computer analysis to identify low-cost concentrator selectors for all practical CPC sizes.

6.4 COLLECTOR PERFORMANCE

6.4.1 Steady State Efficiencies

Concentrators coupled to absorber models were used to identify low cost CPC collectors that are appropriate for electric power generation systems. CPC collector performance for a particular design with characteristics that are listed in Table 6-2, are shown on Figure 6-4 for steady-state conditions at solar noon on September 21. High efficiencies are obtained for temperatures less than 200°C, but thermal losses become large for fluid delivery temperature near 300°C.

6.4.2 Dynamic Performance

The thermal power delivered from the example CPC collector that is identified in Table 6-2, is presented for one-half hourly intervals for winter and summer solstices and spring and fall equinoxes on Figures 6-5 through 6-8. The temperature of fluid delivery was set at 300°C, thus the efficiencies are low. As the steady-state situation in Figure 6-4 indicates, efficiencies for lower temperatures are much higher,

Figure 6-4. Steady-state CPC Efficiency at Noon on September 21

Figure 6-5. Hourly Performance for the CPC Collector on December 21
Outlet Fluid Temperature, 300°C

Figure 6-6. Hourly Performance for the CPC Collector on March 21
Outlet Fluid Temperature, 300°C

Figure 6-7. Hourly Performance for the CPC Collector on June 21
Outlet Fluid Temperature, 300°C

Figure 6-8. Hourly Performance for the CPC Collector on September 21
 Outlet Fluid Temperature, 300°C

Table 6-2. Compound Parabolic Concentrator Example

<u>DESIGN VARIABLE</u>	<u>VALUE</u>
Latitude	33 degrees
Orientation and Tracking	Fixed tilt equal to latitude
Collector Slope	33 degrees
Absorber width	2 m
Absorptance	.95
Emittance	.32
Transmittance	.92
Inlet temperature	40 degrees
Outlet temperature	300 degrees
Concentrator width	2.286 m
Concentrator length	6.096 m
Rim Angle (Acceptance angle)	60 degrees
Reflectance (Reflector surface)	.85
Wind velocity	5 m/s
Ratio of diffuse radiation to direct	.2
Number of CPC's in box	24

however, the set of performance information at 300°C was chosen so that comparisons with other concentrating collectors previously shown can be made at the same fluid temperature.

The thermal losses from the collector clearly dominates, and lowers the collector efficiency. The optimization process considers only minimum cost thermal energy (or power) from optimized collectors so that the collector chosen for illustration in Figures 6-5 through 6-8 is not necessarily an optimum CPC design.

6.4.3 Sensitivity

The thermal energy delivered from a CPC collector is sensitive to the number of units in a module, if there are less than 12 units as shown in Figure 6-9. With a large number of units, there is a greater mass flow rate, and a better heat transfer property compared to a small number of units, low concentration ratio, and low mass flow rate to maintain high fluid temperature from the collector.

Narrow and deep CPC designs have better optical characteristics than wide and shallow reflectors. At times during the year when the radiation is normal to the aperture, there is little effect of acceptance angle variation on the total energy delivered from the collector. However, there is measurable effect at other times because the operating period of the collector is reduced with small acceptance angles as shown on Figure 6-10.

Truncation of a CPC is characterized in this study as an increase in absorber width for the collector module. The "full" CPC produces the greatest amount of energy during a day, as shown in Figure 6-11 (for an absorber width of 1.315 m).

Figure 6-9. Sensitivity of Thermal Energy Delivered to Number of CPC Units in a Module

Figure 6-10. Sensitivity of CPC Collector to Acceptance Angle

Figure 6-11. Sensitivity of CPC Thermal Energy Output to Absorber Width (Constant Acceptance Angle of 35°)

6.5 OPTIMUM COLLECTORS

6.5.1 Factors Included in Analysis

The design variations considered in the optimization analysis for a CPC collector are number of units in a module, effective aperture, collector length and absorber width. Changing optical characteristics with time of day are included in the analysis as well as the thermal properties of the absorber.

6.5.2 Procedure

A matrix of variables and their incremental values is decided, and each combination of design variables is applied to discretized analytic functions to calculate optical and thermal collector performance. Using the cost model with the performance model, collectors which produce energy (or power) at lowest cost are identified. Recognizing that the collector which delivers lowest cost energy may not be the optimum unit in power plant collection field, or the optimum collectors which produce lowest electrical energy costs, a large number of "optimized" collectors are retained in computer memory until the plant optimization is completed.

One of the optimum CPC collectors for an optimum power plant is identified in Table 6-3. A collector module has an aperture width of 1.91 m, and the modules are arranged end to end in the field to form a total length of 1452 m. The CPC reflectors are aligned E-W, concentration ratio is 1.53, and outlet fluid temperature is 150°C. The field array of CPC collectors produce steam more economically than heat in pressurized water.

6.5.3 Results

Optimum CPC collectors are for delivered fluid temperatures less than 200°C. To provide comparison with other line-focus collectors, the

Table 6-3. Design for CPC Collector for an Optimum CPC Power Plant

<u>DESIGN VARIABLE</u>	<u>VALUE</u>
Aperture width of the box	1.91 m
Acceptance angle	35 degrees
Collector length (feederline length)	1452 m
Collector slope	33 degrees
Absorptance (fixed)	.97
Transmittance (fixed)	.95
Emittance (fixed)	.28
Mass Flow Rate	.24 kg/s
Inlet temperature	40°C
Outlet Temperature (Boiling)	150°C
Wind Velocity	5 m/s
Aggregate width of the absorber in the box	1.25 m
Individual absorber width	0.69 m
No. of CPC's in the box	18
Concentration Ratio	1.53
Average Reflectivity	.83

energy cost for a PTC collector is shown on Figure 6-12, for fluid temperature of 100° and 150°C. This is followed by the results of computerized calculations for a CPC collector at 100 and 150°C fluid temperature delivery on Figure 6-13. The CPC and PTC collectors for the same fluid temperature are shown on Figures 6-14 and 6-15. The thermal power costs are significantly greater for the CPC collector for low fluid flow rates and all temperatures.

6.6 MINIMUM COST COLLECTOR FIELDS

The parabolic trough delivering low temperature energy (150°C) and the compound parabolic concentrator were assembled in a field array to determine optimum collector fields. The results are shown in Figure 6-16 where considerable difference in the cost of thermal power is noted. Neither collector exhibits sensitivity to power plant capacity in the range from 1.5 to 10 MW_e.

6.7 MINIMUM COST ELECTRICAL ENERGY

The costs for electrical energy from optimized parabolic trough and the CPC power plants are shown on Figure 6-17. The energy cost from the CPC power plant is 3 to 3.5 times that from a PTC power plant and the electric energy cost from the PTC plant is high at the low steam temperature of 150°C.

Figure 6-12. Optimum PTC Collectors for Low Fluid Temperatures

Figure 6-13. Optimum CPC Collectors for Low Fluid Temperatures

Figure 6-14. Comparison of Optimum Collectors at 100°C Fluid Temperature

Figure 6-15. Comparison of Optimum Collectors at 150°C Fluid Temperature

Figure 6-16. Thermal Power Costs for Optimized CPC and PTC Collector Fields which Deliver Steam at 150°C

Figure 6-17. Cost for Electrical Energy from Optimized CPC and PTC Power Plants at 150°C Steam Temperature

7.0 ORGANIC WORKING FLUIDS

7.1 INTRODUCTION

This section of the report concerns a study of organic fluids that could be suitable for operating Rankine cycle engines for selected power generation subsystems with heat supplied from line-focus solar collectors. Selections of organic fluids were made for different operating ranges in a total temperature range from 100 to 300°C, thermodynamic cycles were analyzed and conceptual turbines were designed. From the conceptual designs, costs were estimated for the selected power plant subsystems which include (1) the turbine generator equipment, (2) condensing and circulating water systems, (3) feed heating systems, (4) cooling towers and, (5) the electrical plant equipment.

7.2 CYCLE FLUID SELECTION

Preliminary screening of acceptable working cycle fluids can be judged with respect to selection parameters developed for the turbine, condenser, and feed heating systems. Such parameters have been examined to assist in cycle fluid selection for automotive and organic cycle waste heat and solar applications, (10), (11), (12).

It is assumed that solar energy is fed into the boiler by means of a circulating transport fluid and coupled to a thermal energy storage facility. There will then be a temperature profile which will depend on the ratio of heat extraction rate to heat capacity flow rate in the transport fluid. If this profile is small (less than 20°C), then a subcritical working fluid, either pure or azeotropic is indicated. If the profile is moderate (20°-100°C), and especially if a similar condenser coolant profile exists, then a subcritical, nonazeotropic mixture should be selected for profile matching. If the profile is

very large, then supercritical heat addition may be favored, as for example, the use of Fluorinol-85 for the 300°C level solar plant.

7.2.1 Turbine Parameters

It is preferable to avoid turbine moisture by choosing a fluid with a near vertical slope on the T-S diagram in the operating range. This means that the I-factor (10) should be near unity.

$$\text{I-factor} = \frac{L}{C_p T} \approx 1. \quad (7-1)$$

(For explanation of symbols, refer to section 7.5). For values much larger than unity, wet expansion will take place. For values much smaller, the expansion will end with the fluid superheated, necessitating a regenerative desuperheater. This may not be disadvantageous if the vapor and liquid specific heats are not too different in that the desuperheater can be used efficiently as a feedwater heating stage.

Turbine size will be minimized by a high energy flux. Assuming that the vapor velocity will be limited to a value proportional to sonic velocity, the mass flux will be proportional to

$$\rho c = \rho \sqrt{\frac{\gamma P}{\rho}} = \sqrt{\frac{\gamma M}{RT}} P, \quad (7-2)$$

approximated by perfect gas laws. Then the net energy flux will be proportional to

$$LP \sqrt{\frac{\gamma M}{RT}}.$$

Since the low pressure end is controlling, $LP_0 \sqrt{\gamma M}$ should be maximized, where P_0 is condenser pressure. This quantity is referred to later as the turbine size parameter.

The working fluid inventory should be approximately inversely proportional to the same quantity.

7.2.2 Condenser Parameter

The condenser surface area will be minimized if the film condensation heat transfer coefficient is maximized. This means that, for a given condenser geometry, the condenser size parameter should be maximized

$$4 \sqrt{\frac{K_L^3 \rho_L^2 L}{\mu_L}}$$

where K_L , ρ_L , μ_L refer to the liquid properties at condenser conditions.

To minimize the cross-sectional flow area in the condenser and piping to the condenser for a given flow rate, or equivalently maximize the cross-sectional energy flux into the condenser for a given size and geometry a fixed allowable pressure drop (ΔP) is assumed.

$$\Delta P = f \frac{L}{D} \frac{W^2}{\rho A^2} \quad (7-3)$$

where f is the friction factor,

$\frac{L}{D}$ is the length to diameter ratio of the passage,

W is the mass flow rate and

A is the passage cross-sectional area.

Rearranging to solve for W^2 and multiplying by L^2 this becomes

$$W^2 L^2 = \frac{\Delta P D \rho A^2}{f L} = \frac{\Delta P D A^2}{f L} \frac{P M}{R T} L^2 \quad (7-4)$$

Noting the pressure drop, friction factor, temperature and geometry are independent of the working fluid, the product WL , an energy transport factor at condenser conditions which should be maximized, becomes:

$$WL \propto L \sqrt{P M} \quad (7-5)$$

Thus the quantity $L \sqrt{P M}$ is estimated at condenser conditions and referred to below as the piping size parameter at condenser conditions.

7.2.3 Feedheating Parameters

Because of irreversibilities requiring regenerative feed heating, it is desirable to reduce the ratio of energy added to the liquid as sensible heat to that added as latent heat. That is

$$\frac{C_p \Delta T}{L},$$

should be minimized. Thus the fluid which maximizes $\frac{L}{C_p}$, which we call the feed-heat parameter, is preferable in this respect.

Also, because of reversibility penalties associated with feed pump work, and subsequent reconversion of this enthalpy in the turbine, it is desirable to minimize the ratio of pump work to latent heat. This requires that $\frac{L \rho g}{P_1}$ be maximized where P_1 is the boiler pressure and approximately equal to the pumping head. This is the feed-pump parameter.

7.2.4 Criteria Summary and Other Properties

In summary, all criteria point to desirability of high molecular weight working fluid. All criteria suggest a high latent heat is desirable except as it relates to dry versus wet expansion in the I-factor, but for many organics $\frac{C_p T}{L}$ will be sufficiently near unity even if a fluid with large latent heat is chosen. The parameters suggest a high condenser pressure is desirable although structural limitations may govern at very high pressures.

Apart from these thermodynamic-related properties, additional items of concern are:

- 1) Long term chemical stability at temperature
- 2) Chemical compatibility with turbine and loop materials
- 3) Compatibility with turbine and feed pump lubricants

- 4) Toxicity, and
- 5) Flammability.

It seems unlikely that any working fluid will be found that is ideal in all respects. Moderate toxicity and flammability may be tolerated provided these materials are routinely handled in quantity for other applications and adequate safeguards are known to exist.

7.2.5 Fluid Selection for the Three Temperature Levels

Although selecting an appropriate working fluid appears to be a difficult task because of the vast number of possible candidates, especially in the lower temperature ranges, preliminary screening on the basis of critical temperature, vapor pressure in the ambient temperature range, and latent heat will eliminate most candidates and often entire classes of compounds. Certain candidates stand out and have been selected for critical screening. One observation is that the hydrocarbons and the halogenated hydrocarbons that are popular as refrigerants generally have very low latent heats and do not compare favorably for use in this application. Some of these standard refrigerants have been included, along with water, for purposes of comparison.

The comparison parameters were calculated from property values in the units indicated in the nomenclature. A base condenser temperature of 25°C was assumed rather than the more appropriate value of 30°C-40°C, because more of the property values were found to be tabulated at that temperature.

7.2.5.1 Choice of Fluid for 100°C Top Temperature

This temperature range is similar to that encountered in many refrigeration situations, and it seems prudent to use a fluid having a history of successful use in this application. A representative

list is given in Table 7-1 showing the values of the selection criteria developed above. Ammonia stands out as superior to the other fluids in terms of the turbine size parameter, the piping size parameter, and the condenser size parameters, while only water excels in terms of the feed heat and feed pump parameters. Even extending the search beyond established refrigerant fluids did not indicate any candidate having properties approaching those of ammonia. The I-factor is slightly larger than desired ($I = 1.5$) indicating that some moisture will be found during expansion, but this will be considerably less than is present with water ($I = 3.4$).

7.2.5.2 Choice of Fluid for 200°C Top Temperature

This temperature range presents a large number of possibilities. Selection parameters were calculated for those compounds listed in Table 7-2 after passing an initial screening. On the basis of known information, ethyl mercaptan appears to be the best. However, not enough is known about this compound, especially its long-term stability, to recommend its use at this time. Overall, methanol is considered to be a choice considering that it is relatively inexpensive and plentiful, and is only slightly soluble in petroleum lubricants.

7.2.5.3 Choice of Fluids for 300°C Top Temperature

At this temperature level, chemical stability becomes important. While many compounds can be identified with suitable physical properties, as listed in Table 7-3, it is difficult to find information with regard to long term stability. Pyridine and pyridine-water near azeotropic proportions appear to be superior to most other compounds. These are being considered for other Rankine cycle applications (13). The pyridine-water mixture is recommended at about 30 mole percent pyridine.

Table 7-1. Properties of Fluids for 100°C Temperature

Compound	Molecular Weight	Crit. Temp. °C	Vapor Pressure at 25°C mm-Hg	I-Factor at 50°C	Turbine Size Parameter at 25°C ($\times 10^{-9}$)	Piping Size Parameter at 25°C ($\times 10^{-7}$)	Condenser Parameter at 25°C	Feed-Heat Parameter at 25°C	Feed-Pump Parameter at 25°C ($\times 10^{-4}$)
Water-H ₂ O	18.015	374	23.69	3.38	.276	5.02	1180	582	377
Ammonia-NH ₃	17.03	133	7518	1.50	41.1	41.7	1690	247	2.51
Sulfur Dioxide SO ₂	64.06	157	1142	1.27	9.0	14.7			2.18
Butane-C ₄ H ₁₀	58.12	152	1823	.57	5.19	11.8	146	163	1.86
R-114 CClF ₂ -CClF ₂	170.9	159	1620	.53	2.85	6.95	162	128	1.97

Table 7-2. Properties of Fluids for 200°C Top Temperature

Compound	Molecular Weight	Crit. Temp. °C	Vapor Pressure at 25°C mm-Hg	1-factor at 50°C	Turbine Size Parameter at 25°C ($\times 10^{-8}$)	Piping Size Parameter at 25°C ($\times 10^{-7}$)	Condenser Parameter at 25°C	Feed-Heat Parameter at 25°C	Feed-Pump Parameter at 25°C ($\times 10^{-4}$)
Water-H ₂ O	18.015	374	23.69	3.38	2.76	5.02	1180	582	20.9
Methanol-CH ₃ -OH	32.04	240	126	.90	8.86	7.61	658	475	3.14
Acetone CH ₃ COCH ₃	58.08	235	230	.94*	9.71	6.40	430	226	2.05
Carbon Disulfide-CS ₂	94.2	273	362	1.27	14.3	5.98	449	369	2.09
Ethanol C ₂ H ₆ O	46.1	243	58.7	1.63	3.81	4.85	304	326	3.22
Isopropyl Alcohol C ₃ H ₈ O	60.1	235	32.5	1.36	2.43	4.10	377	358	
Ethyl Mercaptan C ₂ H ₆ S	62.1	231	527	.88*	19.5	6.28			
Diethyl Amine C ₄ H ₁₁ N	73.1	223	235	.68*	8.79	5.44			.67
Hexane C ₆ H ₁₄	86.2	235	151	.59*	5.46	4.18	299	166	1.80
R-30 CH ₂ Cl ₂	84.9	249	42.2	1.15*	1.38	1.42	249		
R-113 CCl ₂ CClF ₂	187.4	214	33.5	.67	0.71	1.21	172	172	7.24

*Based on estimated values

Table 7-3. Properties of Fluids for 300°C Top Temperature

Compound	Molecular Weight	Crit. Temp. °C	Vapor Pressure at 25°C mm-Hg	f-factor at 50°C	Turbine Size Parameter at 25°C ($\times 10^{-7}$)	Piping Size Parameter at 25°C ($\times 10^{-7}$)	Condenser Parameter at 25°C	Feed-Heat Parameter at 25°C	Feed-Pump Parameter at 25°C ($\times 10^{-4}$)
Water-H ₂ O	18.015	374	23.69	2.94	27.6	5.02	1180	582	3.73
Pyridine C ₅ H ₅ N	79.1	340	24.5	1.04	12.1	2.47	628	334	2.05
Pyridine 30% Water 70% Mole Fractions	36.3	325	37.4	1.26	19.2	3.14	618	332	1.00
Tetrachloro- ethylene C ₂ Cl ₄	165.8	350	18.5	1.90	6.15	1.34	397		
Dimethyl Disulfide C ₂ H ₆ S ₂	94.2	333	28.6	.90	11.9	2.13			
n-Nonane C ₉ H ₂₀	128.2	322	4.3	.42	1.80		167		
CP-32 Nonsanto Data	79.1	347	21.1	.75	9.62	1.88	395	294	1.92

In addition to a subcritical fluid, a supercritical fluid, Fluorinol-85, has been selected for analysis at a supercritical pressure because investigation determined that it was suitable for heat extraction with a large temperature profile using segmented or stratified thermal energy storage.

7.3 THERMODYNAMIC CYCLE ANALYSIS AND CONCEPTUAL APPARATUS DESIGN

Thermodynamic cycle analyses were completed for the four selected working fluids, (1) ammonia, (2) methanol, (3) 75 pyridine/25 water (by weight) and (4) 85 Fluorinol/15 water, and for three selected power levels, 3 MWe, 30 MWe, and 300 MWe. The calculations were performed using appropriate temperature-entropy (T-S) diagrams and with and without vapor extraction for feed heating. The Fluorinol-85 analysis was a supercritical fluid analysis. Temperature-entropy diagrams were available for ammonia (14), 75 pyridine/25 water (15), and Fluorinol-85 (16). The T-S diagram for methanol was generated from the available thermodynamic information shown in Appendix B, and it is incorporated with the cycle analysis.

A computer program was written to calculate the turbine size, number of stages, and stage efficiency. The flow was treated one-dimensionally with all parameters referred to the mean blade radius. Two blading types were considered: impulse for the high pressure stages; and 50% reaction for the low pressure stages. The turbine efficiency was estimated from Baljé's correlation (16), and included as an iterated input to the turbine performance calculation. An allowance for gear losses was included in the 3 MWe case where these losses were significant. No allowance was made for generator or auxiliary power losses.

Efficiencies were calculated for each working fluid at the three power levels considered over a range of peak temperatures to determine the extent of efficiency variation with peak temperature. Once these variations were known curves bearing a consistent relationship to the Carnot efficiency curve were drawn in as shown in Figure 7-1. This figure shows the efficiency plot for the case where there is no regenerative feed heating and therefore feed enters the boiler at the 45°C condenser outlet temperature and this implies a collector coolant return temperature of about 50°C. The curves are considered to represent efficiencies that could be obtained using the "best" organic working fluid for each temperature. This "best" fluid while it may not exist is useful in conducting a survey study such as the present one. The data of Figure 7-1 indicates that methanol actually yields poorer performance than would be expected from the curve fit through the ammonia and pyridine/water results. It also shows that decreases in the output power level result in efficiency degradations only at the higher peak temperatures. The best efficiencies that can reasonably be expected using organic working fluids at a given peak design temperature are then those given by the heavy lines of Figure 7-1 for the case where feed heating is not employed.

When regenerative feed heating is used in a cycle one observes an increase in cycle efficiency primarily at the higher peak temperature levels. The extent of this improvement is indicated by the curves shown in Figure 7-2 which were obtained in the same manner as those of Figure 7-1. The feed from the condenser was supplied to the lowest temperature feed heater at 45°C to obtain the results of Figure 7-2. The temperature at which the feed would be supplied to the collector

Figure 7-1. Cycle Efficiencies without Feed Heating

Figure 7-2. Cycle Efficiencies with Feed Heating

Figure 7-3. Feed Return Temperatures for Feed Heating

coolant-fired boilers after the feed heating stage is shown in Figure 7-3. The temperature of the coolant returning to the collector system would be expected to be about 5°C above these values for a properly designed unit.

Comparison of Figures 7-1 and 7-2 shows the efficiency of the ammonia system is not enhanced through the use of feed heating. This occurs because the low peak temperature results in the need for only one feed heating stage. Cycle efficiency is however enhanced substantially through the use of feed heaters for the methanol and the pyridine/water systems.

The Fluorinol-85 system was evaluated only at the 300°C peak temperature condition where it is supercritical. The resultant efficiency data should facilitate comparison of a supercritical fluid with the water and pyridine/water systems. Because Fluorinol-85 is supercritical at this condition it requires no feed heating and the return temperature of the collector coolant would be about 5°C above the 45°C condensate return temperature. The following efficiencies were calculated for Fluorinol-85 at the power levels considered.

<u>Power</u>	<u>Turbine Efficiency</u>	<u>Cycle Efficiency</u>
3 MW	83.8	29.2
30 MW	88.5	30.8
300 MW	88.6	30.9

The computer program for cycle analysis was enlarged to estimate performance of the power plant at off-design power levels. The off-design turbine characteristics were calculated in considerable detail for the methanol cases, since the cycle analysis had been programmed

with the developed T-S data. A least squares functional fit of the Baljé turbine efficiency diagram was added to the program with provision for iteration on the cycle until the blade group efficiencies used in the analysis agreed with the Baljé efficiencies. It was found that the off-design cycle efficiency could be correlated by the equation:

$$\frac{\eta}{\eta_o} = \left(\frac{P_p}{P_f}\right)^{0.0898} \left[-0.3821 + 2.5774 \frac{\eta_c}{\eta_{co}} - 1.1953 \left(\frac{\eta_c}{\eta_{co}}\right)^2 \right] \quad (7-6)$$

where:

η_{co} is the Carnot efficiency based design temperatures (i.e. 45°C condensate and the peak temperatures) and power level,

η_o is the efficiency at rated conditions given by Figure 7-1 or 7-2, and

η_c is the Carnot efficiency at off-design condenser and/or peak temperature, and

(P_p/P_f) is the ratio of partial-to-design full power level.

Equation (7-6) can be rearranged to express the relation between partial-to-full power output ratio (P_p/P_f) and partial-to-full thermal energy input ratio (Q_p/Q_f) :

$$\left(\frac{P_p}{P_f}\right)^{0.9102} = \left(\frac{Q_p}{Q_f}\right) \left[-0.3821 + 2.5774 \frac{\eta_c}{\eta_{co}} - 1.1953 \left(\frac{\eta_c}{\eta_{co}}\right)^2 \right] \quad (7-7)$$

Equation (7-7) may be considered valid for those conditions not leading to greater volumetric flow rates than those present at design conditions. Mach number limitations would enter slightly beyond design conditions.

If the power plant is to be operated at constant turbine inlet and 45°C condenser outlet temperature conditions but at a part-load power level, the preceeding equations reduce to:

$$\eta = \eta_o \left(\frac{P_p}{P_f}\right)^{0.0898} \quad (7-8)$$

and

$$\frac{Q_p}{Q_f} = \left(\frac{P}{P_f}\right)^{0.9102} \quad (7-9)$$

If the plant is to be operated such that constant volumetric flow conditions are maintained at the turbine, the equations considered to be applicable (which were also derived from the cycle data for methanol) are

$$\eta = \eta_o \left(\frac{P}{P_f}\right)^{0.2174} \quad (7-10)$$

and

$$\frac{Q_p}{Q_f} = \left(\frac{P}{P_f}\right)^{0.7826} \quad (7-11)$$

Since the computer programs to analyze the Fluorinol-85, ammonia, and Pyridine-75 cases were not available (T-S diagrams were used), it was not possible to perform as detailed off-design analyses for these cycles. However, several points were checked, and it appears that the above normalized equations are adequate for all cycle fluids studied.

To determine the throttle or peak temperature (T_p) which would be observed for a plant designed to operate at a thermal input power level (Q_f) but operating at Q_p , and for a given collector coolant temperature entering the boiler (T_H), some mathematical description of the boiler is required. This heat exchanger is adequately described by

$$T_p = T_H - TTD_B \left(\frac{Q_p}{Q_f}\right), \quad (7-12)$$

where TTD_B is the temperature difference across the heat exchanger at rated power and is assumed to be equal to about 10°C for a properly designed unit. The temperature drop of the collector coolant (ΔT_R)

having a specific heat, C_p , associated with its passage through the heat exchanger at a mass flow rate ($\dot{m}_R$) is

$$\Delta T_R = Q_p / \dot{m}_R C_R , \quad (7-13)$$

where $\dot{m}_R$ must be sufficiently high so that the temperature of collector coolant exiting the heat exchanger is several degrees greater than the boiler feed inlet temperature, as given by Figure 7-3, for the case where feed heat is being used. When a plant is designed to operate without feed heating a second heat exchanger would be required to raise the feed temperature to within about 10°C of the boiler (peak) temperature. This would be accomplished by circulating the collector coolant exiting the boiler through this second heat exchanger. The temperature difference between the collector coolant entering this unit and the cold working fluid from the condenser at 45°C would be given by

$$\delta T_{HX} = T_H - \Delta T_R - 45 . \quad (7-14)$$

The temperature drop of the collector coolant $\Delta T_R'$ as it passes through this preheating unit would be equal to

$$\Delta T_R' = \delta T_{HX} - 10 \quad (7-15)$$

It would also be possible to boil the organic working fluid in the collectors themselves and although this would require a large working fluid inventory the heat exchangers could be eliminated.

7.4 COSTING OF APPARATUS

The costing of the power plant apparatus has been separated into five accounts:

- Turbine-generator equipment
- Condensing system and circulating water system
- Feed heating system
- Cooling towers
- Electrical plant equipment

The estimates for these accounts are summarized below:

7.4.1 Turbine-Generator Equipment

The cost for the turbine-generator can be estimated from the cost equation of the type developed in the previous NSF funded study on the water/steam cycle (3).

$$C_{TG} = \alpha P_A^{0.75} + \beta E_A, \quad (7-16)$$

where α , β depend on working fluid type

C_{TG} = cost of turbogenerator equipment

P_A = power in kilowatts

E_A = exhaust end area in m^2 (given in Figure 7-4)

<u>Fluid</u>	<u>α (\$/Kw)</u>	<u>β (\$/m²)</u>
Water	710	140,000
Fluorinol-85		
Pyridine-75		
Methanol		
Ammonia	355	140,000

The pricing formula used for water has been estimated to apply to the Fluorinol-85, Pyridine-75, and Methanol turbines, because of the similarity in turbine sizes. However, the ammonia unit is considerably smaller, and an estimate of its cost checks reasonably well with the published pricing given by Sluzarek (14) for ammonia units.

Figure 7-4. Power Levels and Exhaust Areas for Selected Working Fluids

7.4.2 Condensing System and Circulating Water System

The condensing and circulating water systems are described by the equations developed in the previous CSU/Westinghouse study (3).

$$S_A = \frac{Q_R}{\delta \Delta T_R} \ln \left\{ \frac{TTD + \Delta T_R}{TTD} \right\} \quad (7-17)$$

$$\frac{C_C}{Q_R} = \left[\frac{10^5}{Q_R} + \frac{\epsilon}{\Delta T_R} \ln \left\{ \frac{TTD + \Delta T_R}{TTD} \right\} \right] \quad (7-18)$$

$$C_C = 10^5 + \epsilon S_A \quad (7-19)$$

The above equations were developed by coupling condenser design and pricing information with prices of the balance of the condenser and circulating water systems (18). The condensing heat transfer coefficients can be calculated for the four fluids plus water with the transport properties and condenser design procedure known.

Fluid	Thermal Conductivity (W/m C) (K_L)	Viscosity (centipoise) (μ_L)	Density (Kg/m ³) (ρ_L)	Latent Heat (joules/Kg) (L)	$\left[\frac{K_L^3 \rho_L^2}{\mu_L} \right]^{1/4}$
Water	0.593	0.85	1000	.967	1.000
Fluorinol-85	0.129	1.43	1350	.177	0.188
Pyridine-75	0.221	1.50	1000	.272	0.216
Methanol	0.215	0.50	770	.469	0.445
Ammonia	0.502	0.10	590	.469	0.980

The above physical properties were derived from several references, (12), (13), (18). The group $\left[\frac{K_L^3 \rho_L^2}{\mu_L} \right]^{1/4}$ yields the variation in condensing heat transfer coefficient and (U) can be calculated from condenser design procedures and also the values of δ and ϵ .

<u>Fluid</u>	<u>U-Value</u> <u>(kW/m²°C)</u>	<u>δ</u>	<u>ε</u>
Water	3.41	3.40	64.6
Fluorinol-85	0.87	0.89	115.1
Pyridine-75	0.98	0.97	115.1
Methanol	1.65	1.64	83.9
Ammonia	3.38	3.37	64.6

The pricing equation for the condenser and circulating water system contains two price contributions: the condenser itself, and the balance of the systems such as pipes, pumps, valves, etc. (19). Referred to the water/steam system (3), these contributions are given by

$$C_C^W = 10^5 + 61.3 S_A^W + 158.2 S_A^W. \quad (7-20)$$

It is unlikely that much change will be observed in the price term $158.2 S_A^W$, when the cycle fluid is changed and with it the U-value. With unchanged values of heat exchanger temperature differences, the condenser area will vary inversely with the U-value, so that the condenser price will increase with S_A , but the balance of the system referred to S_A will have a different multiplier. The determination of ζ gives the following results.

<u>Fluid</u>	<u>U-Value</u> <u>(kW/m²°C)</u>	<u>158.2 ($\frac{U}{3.41}$)</u>	<u>ζ (\$/m²)</u>
Water	3.41	158.2	219.5
Fluorinol-85	0.87	46.3	102.2
Pyridine-75	0.98	39.8	107.6
Methanol	1.65	76.4	137.7
Ammonia	3.38	157.1	218.4

7.4.3 Feed Heating System

The feed heating system cost is estimated as a factor varying with the cost of the condenser/circulating water system for each of the cycle fluids.

$$C_{fh} = \theta C_c, \quad (7-21)$$

where C_{fh} is the cost of feed heating system

θ is the ratio of feed heating system cost to condensing/circulation water system.

The value of θ is estimated by considering changes in feed heater cost associated with changed U-value, changes in pump cost due to changed pump size and fluid flow, changes in piping, etc. cost associated with changes in the number of extractions (and feed heaters), and changes in the ratio of cost of the feed heating system to cost of the condenser/circulation water system associated with changes in the ratio of feed flow rate to condensate flow rate. These changes are estimated with cycle data of this study coupled with equipment/cost data of the study by United Engineers and Constructors, Inc. In addition, cost changes with size of plant, as approximated here, have been compared with data developed over a 200 MWe to 600 MWe study completed and updated by Westinghouse in 1965 and 1970 (19).

<u>Fluid</u>	<u>Condensate Rate*</u>	<u>Plant Factors</u>		<u>No. Feeds</u>	<u>θ</u>
		<u>Feed Rate*</u>	<u>Peak Temperature</u>		
Water	0.67	0.33	300°C	6	1.0
Fluorinol-85	1.00	0.00	300°C	0	0.0**
Pyridine-75	0.33	0.67	300°C	5	6.1
Methanol	0.65	0.35	200°C	4	1.0
Ammonia	0.88	0.12	100°C	1	0.2

* Relative rates summing to 1.0

** Supercritical cycle with no feed heating

7.4.4 Cooling System

The cooling system is unchanged from that described in the earlier study on water/steam (3) and its cost estimate is unchanged from that given in that study.

7.4.5 Electrical Plant Equipment

The price of the electrical plant equipment for the plant is approximated by

$$C_e \approx 10^{-2} P_f^{0.5} \quad (7-22)$$

This correlation is derived from data in two references (17), (19).

7.4.6 Heat Exchangers

Two heat exchangers have been developed for adding heat from the solar collector coolant to the power plant loop. The first is the boiler and the second is the preheater which is required when feed heating is not used.

Two equations pertain to the boiler, the first of which is

$$S_H = \frac{Q_p}{4.53 \Delta T_R} \ln \frac{TTD_B + \Delta T_R}{TTD_B}, \quad (7-23)$$

where

U-Value is $4.53 \text{ KW/m}^2\text{C}$,

S_H is boiler heat transfer area,

Q_p is boiler heat rate,

ΔT_R is the boiler range and

TTD_B is the terminal temperature difference for the boiler.

A numerical value of 0.257 is developed from an in-tube convective heat transfer coefficient of $(0.16)^{-1} \text{ m}^2\text{C/KW}$ which corresponds to boiling coefficients of approximately 85 to $142 \text{ KW/m}^2\text{C}$, and are

values developed from the Cichelli and Bonilla curves in McAdams (1954 edition, p. 384). Application of fouling factors and tube metal thermal resistances, taken from Westinghouse design charts, reduced the U-Value for all four fluids to approximately $4.54 \text{ kW/m}^2\text{°C}$. A second equation develops the cost of this boiler at pressures of 70 to 160 atmospheres using 18 gauge SS 304 tubing in the following form:

$$C_H^W = 70,000 + 98.7 S_H, \quad (7-24)$$

where C_H^W is the cost of the boiler, not installed.

The feed from the condenser must be heated in a heat exchanger whose operation is approximately described by

$$S_{HX} = \left(\frac{Q_{HX}}{U_{HX} \delta T_{HX}} \right) \left(1 - \frac{\Delta T_R^1}{\delta T_{HX}} \right)^{-1}, \quad (7-25)$$

where S_{HX} is the area of the heat exchanger,

Q_{HX} is the heat rate of the heat exchanger,

U_{HX} is the U-value of the heat exchanger, and

δT_{HX} is the temperature difference between the hot fluid in and the cold fluid in at 45°C . The U-values depend on the fluid as follows:

Fluid	U-Value $\text{kW}/(\text{m}^2\text{°C})$
Water	3.12
P-75	1.56
F-85	1.25
Methanol	2.21
Ammonia	3.97

The cost of this heat exchanger with high carbon steel tubing for 70 to 130 atmospheres pressure rating is given by

$$C_{HX}^N = 70,000 + 76.8 S_{HX} , \quad (7-26)$$

where C_{HX}^W is the cost of the uninstalled heat exchanger.

7.4.7 Power Plant Piping for the Heat Exchangers

Piping cost is best expressed as an adder to the condenser and circulating water system cost as given in Reference (3) and the C_C^W factor of that report is now expressed as

$$C_C^W = 10^5 + \delta S_A , \quad (7-27)$$

where δ is increased in value to account for the rest of the power loop piping costs, and shown below:

<u>Fluid</u>	<u>δ \$/m²</u>
Water	272.2
P-75	141.0
P-85	135.6
Methanol	181.8
Ammonia	317.4

7.4.8 Cooling System

The cooling system is unchanged from that described in the earlier study on water/steam (3) and the cost estimate is unchanged from that given previously

$$R_{CT} = 5200 \Delta T_A^{-0.4} T_W^{0.233} T_O^{-2} r_a + 97 \left(\frac{0.048}{\Delta T_R} + 0.002 \right) . \quad (7-28)$$

7.4.9 Electrical Plant Equipment

The price of the electrical plant equipment for the plant is approximated by

$$C_e \approx 10^{-2} P_f^{0.5} . \quad (7-29)$$

This correlation is derived from data in two references, (18) and (20).

7.5 NOMENCLATURE

- A - piping cross-sectional area, cm^2
- c - velocity of sound, cm/s
- C_c - price of condenser and circulating water system, \$
- C_c^w - price of steam condenser and circulating water system, \$
- C_e - price of electrical plant equipment, \$
- C_{fh} - price of feed heating system, \$
- C_{TG} - price of turbogenerator, \$
- C_L - specific heat of liquid, $\text{J}/(\text{kg}\cdot^\circ\text{C})$
- C_p - specific heat at constant pressure of vapor, $\text{J}/(\text{kg}\cdot^\circ\text{C})$
- C_R - specific heat of collector coolant, $\text{J}/(\text{kg}\cdot^\circ\text{C})$
- D - pipe diameter, cm
- E_A - exhaust area of turbine, m^2
- f - friction factor
- K_L - liquid thermal conductivity, $\text{W}/(\text{m}\cdot^\circ\text{C})$
- L - latent heat of vaporization, J/kg
- l - pipe length, cm
- $\dot{m}_R$ - collector coolant flow rate through boiler, kg/s
- M - molecular weight
- P - vapor pressure, mm-Hg
- P_A - electric power, kW
- P_f - full load electric power, kW
- P_o - vapor pressure at condenser, mm-Hg
- P_p - part-load electric power, kW
- P_1 - vapor pressure at boiler, mm-Hg
- Q_f - full load thermal power, kW
- Q_p - part load-thermal power, kW

- Q_R - condenser heat rejection rate, kW
- R - gas law constant
- R_{CT} - annualized price of cooling tower, \$/yr
- r_a - annualized rate of amortization, per year
- S_A - condenser surface area, m^2
- S_A^w - steam condenser surface area, m^2
- T - temperature, $^{\circ}K$
- T_o - condenser outlet temperature, $^{\circ}C$
- T_H - collector coolant inlet temperature to boiler - $^{\circ}C$
- T_p - peak temperature in heat engine cycle - $^{\circ}C$
- ΔT_A - cooling tower approach to wet bulb, $^{\circ}C$
- ΔT_R - boiler range, $^{\circ}C$
- $\Delta T_R'$ - feed heat exchanger range, $^{\circ}C$
- TTD - terminal temperature difference, $^{\circ}C$
- TTD_B - terminal temperature difference in boiler, $^{\circ}C$
- U - overall heat transfer coefficient, $kW/(m^2 \cdot ^{\circ}C)$
- w - mass flow rate, g/s
- α - coefficient of turbogenerator pricing equation
- β - coefficient of turbogenerator pricing equation
- γ - ratio of specific heats
- δ - heat transfer parameter, $kW/(m^2 \cdot ^{\circ}C)$
- δ_{HX} - temperature difference between inlet temperatures to feed heat exchanger, $^{\circ}C$
- ϵ - price heat transfer parameter, $(\$ \cdot ^{\circ}C)/kW$
- ξ - price parameter, $\$/m^2$
- η - off-design cycle efficiency
- η_o - cycle efficiency at design power and temperature

- η_c - Carnot efficiency, based on off-design temperatures
- η_{co} - Carnot efficiency, based on rated temperatures
- θ - cost parameter, C_{fh}/C_c
- μ_L - liquid viscosity, cp
- ρ - vapor density, kg/m^3
- ρ_L - liquid density, kg/m^3

8.0 STORAGE CONCEPTS FOR ORGANIC FLUIDS

8.1 INTRODUCTION

The storage of heat energy for systems employing organic fluids, either in the transfer loop or in the working fluid loop, offers many different potential possibilities. At first glance, storing heat in an organic fluid appears attractive in that the cost of the storage vessel is much less for a low vapor pressure organic fluid than that of a high pressure vessel for water. In general, organic fluids are relatively expensive and usually have specific heats lower than water. The result is that the cost of storing heat in the organic fluid itself becomes very expensive.

As an example, if one considers a typical heat transfer fluid such as Therminol 66, which has a specific heat of about $0.5 \text{ cal/(g}\cdot^{\circ}\text{C)}$ in the normal operating temperature range, the cost of storing energy can be obtained by noting that this specific heat is equal to $2.26 \times 10^3 \text{ W/(s}\cdot\text{kg)}$, or $6.27 \times 10^{-4} \text{ kWh/kg}$. The cost of storing energy in organic fluids is 1600 times the cost of materials in dollars per kilogram divided by the temperature difference employed. An examination of tank costs reveals that the tank component is negligible compared to the cost of the material. If a temperature difference of 100°C is used, the costing rate is approximately $\$16/\text{kWh}$ of thermal energy stored. By way of comparison, steam accumulators at temperatures of 300°C have a costing rate of $\$10/\text{kWh}_t$, and if lower temperatures are considered, such as 200°C , the costing for steam accumulators decrease to about $\$2/\text{kWh}_t$.

There are five basic components involved in the solar thermal electric system that involve the choice of storage concepts and working fluids. These are: (1) Collectors, (2) Transport, (3) Storage,

(4) Exchanger, and (5) Heat engine. It is possible to integrate the heat exchanger with storage, in which case four primary subsystems are involved. In the collector itself, it is possible to have pressurized water operation, boiling water operation, or a heat transfer fluid or working fluid, which is vaporized in the collector. The transport fluid in general, although not necessarily, will be the same as that used in cooling the collector absorber surface. Storage may be either in the transport fluid, the working fluid, a third fluid, or in a salt or some material where heat of fusion can be used. The working fluid in each case will be considered as an organic fluid.

There are numerous combinations which can lead to a satisfactorily working system; however, some choices can be extremely expensive and it is necessary to evaluate the cost factors in at least an approximate manner to determine which concepts are economical and which should be rejected on the basis of cost.

8.2 HEAT STORAGE CONCEPTS

Possible conceptual arrangements for thermal energy storage with organic working fluids are shown in Figure 8-1a through f. Figure 8-1a shows a system with common transfer and storage fluid. A separate heat exchanger or vaporizer is needed for the organic working fluid. The transfer fluid can be either pressurized water or an organic fluid. A similar but slightly different arrangement is shown in Figure 8-1b, where the vaporizer is contained within the storage shell. A third type of system, shown in Figure 8-1c, involves a separate storage fluid and there is the possibility of using three different fluids: a transfer fluid, a storage fluid, and a working fluid.

Figure 8-1 (a-c). Conceptual Thermal Storage Arrangements

Figure 3-1 (d-f). Conceptual Thermal Storage Arrangements

The arrangement shown in Figure 8-1d introduces the concept of heat storage in salt or some other material where the heat of fusion of the material increases the thermal energy storage density. There are numerous types of materials available, some of which are listed in Table 8-1, along with their pertinent properties.

The choice of storage material is extremely difficult and a good selection cannot be made since there are many factors in addition to the specific heat, density, and cost of the materials which must be considered. One factor, which is rather difficult to assess, is that many salts can become supercooled; i.e., the heat of fusion is not available until the salt drops to a temperature appreciably below the normal fusion temperature. This can be alleviated in some cases by proper seeding or other arrangements in the storage unit. An additional factor which is difficult to assess is the corrosiveness of many of the materials involved. This includes not only the molten salts but also the potential for molten metals to react with the container walls, and as a result, either weaken the vessel or destroy it. Systems have been developed for storing molten metals and molten salts at high temperatures, and although problems are anticipated in earlier designs, it is believed that solutions can be obtained which will permit the storage of heat in materials which change state as they are charged and discharged.

In addition to the possibility of supercooling, some heat storage materials develop a "memory" i.e., when charged and discharged in a specific manner, a subsequent change in the manner or rate of charge or discharge results in behavior that is difficult to predict. An additional and very important factor in the design of storage systems is that in the solid state many of the storage materials have relatively low thermal

Table 8-1. Thermal Energy Storage Utilizing Heat of Fusion for
Some Materials in the Range, 0° to 400°C

Material	Formula	Melting Point °C	Heat of fusion		Density g/cm ³		kWh/m ³
			cal/g	W·s/kg x 10 ⁻⁵	solid	liquid	
Ice	H ₂ O	0	80	3.3			
Calcium chloride	CaCl ₂ ·6H ₂ O	29-39	41.6	1.7	1.7		82.2
Lithium nitrate	LiNO ₃ ·3H ₂ O	29.9	70.7	2.9			
Sodium sulphate	Na ₂ SO ₄ ·10H ₂ O	32	57.7	2.4	1.46		97.9
Sodium carbonate	Na ₂ CO ₃ ·10H ₂ O	32-36	63.8	2.6	1.44		106
Sodium orthophosphate mono H	Na ₂ HPO ₄ ·12H ₂ O	34.6	66.8	2.8	1.5		116
Calcium nitrate	Ca(NO ₃) ₂ ·4H ₂ O	41	50	2.1	1.8		105
Sodium	Na	98	27.4	1.1	0.9		28.6
Lithium	Li	180	150	6.3	0.5		87.2
Aluminum chloride	Al ₂ Cl ₆	192	63.6	2.7	2.44	1.3	177
Lithium nitrate	LiNO ₃	250	87.8	3.6	2.38		243
Sodium chlorate	NaNO ₃	255	49.7	2.1	2.49		144
Sodium nitrate	NaNO ₃	310	44.2	1.8	2.26		116
Sodium hydroxide	NaOH	320	50	2.1	2.13		123
50% LiF - 50% BeF ₂		360	153	6.4	2.6, 1.9		391
Potassium hydroxide	KOH	360	35	1.5	2.0		81

Note: 1 g cal x 4,186.8 = 1 W·s/kg

1 g cal x 1.163 x 10⁻³ = 1 kWh/kg

conductivity, and as a result, careful design procedures are required if power is to be withdrawn from the unit at a reasonable rate without involving an exceptionally large temperature drop.

The storage system shown in Figure 8-1e is applicable to those cases where the working fluid vapor pressure is lower than the transfer fluid vapor pressure. In this example, a fluid such as steam would flow through small diameter tubes which are surrounded by larger tubes containing salt. The working fluid then flows on the outside of the tubes containing the salt. One of the advantages of this system is that the portion of the salt which freezes first is the large diameter, and as a result the temperature drop is smaller than would occur if the cold side were the inside of the tubes.

The working and transfer fluids may be interchanged if a working fluid having an appreciably larger vapor pressure than the transfer fluid is used. This would allow for the cost of the large container to be appreciably smaller in both cases. However, in the latter case (working fluid inside the tubes), the advantage of maintaining a low temperature difference is not obtained. For cases where both the transfer fluid and working fluid vapor pressures are high, a storage system such as shown in Figure 8-1f is possible. In this case, both the transfer fluid and working fluid are passed through the storage volume in a counter flow manner, the transfer fluid tubes and working fluid tubes are interspersed, and the whole volume surrounding them is filled with a storage material having a large heat of fusion.

Numerous other types of storage concepts can be developed; however, it is believed that a review of some of the heat transfer characteristics

involved with the various types of fluids and solids should be examined before any further consideration is given to which type of systems appear best.

8.3 HEAT TRANSFER RELATIONS FOR STORAGE SYSTEMS

The film coefficients involved for heat transfer between tubular surfaces is considered in detail by McAdams (23) and Babcock and Wilcox (24).

In a detailed heat exchanger design there are many factors involved. Only the simple relations are described below for individual pipe heat transfer coefficients applicable to longitudinal flow, typical of fluids inside the pipe, and for cross flow, typical for fluids flowing over the pipes.

The power flow for heat transfer from a fluid to a pipe wall surface, is given by the relation

$$P' = C_f \Delta T \quad (8-1)$$

where P' = the power density, W/m^2 ,

C_f is the surface film coefficient, $W/(m^2 \cdot ^\circ C)$

ΔT is the temperature difference, $^\circ C$.

For the case of longitudinal flow, the film coefficient is

$$C_{fL} = k_{fL} v^{0.8} d^{-0.2} \quad (8-2)$$

where C_{fL} refers to longitudinal flow,

k_{fL} is a constant for a given type of fluid at a specified temperature,

v is the fluid velocity, m/s

d is the tube diameter, m .

For the longitudinal flow case, the constant is

$$k_{fl} = 2.1 \times 10^{-2} c_p^{0.4} \sigma^{0.6} \mu^{-0.4} \delta^{0.8} \quad (8-3)$$

where c_p is the specific heat, W-s/kg °C

σ is the thermal conductivity, W/(m·°C),

μ is the absolute viscosity, kg/(m·s)

δ is the material density, kg/m³.

For the case of cross flow where the fluid is flowing over the pipe, the heat transfer film coefficient is

$$C_{fc} = k_{fc} v^{0.6} d^{-0.4} \quad (8-4)$$

where, in this case, the constant for cross flow, k_{fc} , is

$$k_{fc} = 0.23 c_p^{0.33} \sigma^{0.67} \mu^{-0.28} \delta^{0.6} \quad (8-5)$$

It should be noted that different exponents are used in determining the film coefficient and also the influence of fluid velocity and tube diameter.

Typical values for the heat transfer film coefficients for a number of different materials are given in Table 8-2. An examination indicates, for example, transfer of heat energy is more effective with water than for air, or steam. In addition, the transfer coefficients for superheated steam are less than for saturated steam. The values shown for superheated steam were obtained from reference (24). In general the organic fluids have lower heat transfer film coefficients than water.

The transfer of heat from a liquid core through a solid frozen section of pipe to the pipe surface is given by the relation:

$$P/l = 2.73 \sigma \Delta T / \text{Log } (d_2/d_1) \quad (8-6)$$

Table 8-2. Heat Transfer Film Coefficient Constants

Material	Temp. °C	k_{ft}			k_{fc}
Air 1 atm	100	3.5			4.8
Air 1 atm	600				7.2
Water	100	3×10^3			2×10^3
Steam	p(atm)	<u>sat</u>	<u>30</u>	<u>100</u>	<u>200</u>
	175	6.5			
	250	9.2	7.3		
	300	13	6.7		
	350	24	6.4	9.4	
	500		6.9	7.7	9.4
	600		7.3	7.8	8.7
Therminol 66	200	1.5×10^3			1.1×10^3
Methanol (liquid)	100	1×10^3			616
Methanol (vapor)					
Fluorinol 85 (liq)	30	5.7×10^2			362
Ammonia (liquid)	30	2.9×10^3			1.5×10^3

where P/l is the power transfer, W/m of length of the cylindrical tubular structure,

σ is the thermal conductivity, W/(m·°C) of the solidified salt or heat of fusion material,

ΔT is the temperature difference between the unfrozen core and the wall of the cylindrical container, °C,

d_2 and d_1 are respectively the inside diameter of the container and the diameter of the molten core, m.

Equation 8-6 can be solved for the temperature drop across the solid portion of the material as a function of the power transfer per meter length of the tubular container,

$$\Delta T/(P/l) = \text{Log } (d_2/d_1)/2.73\sigma . \quad (8-7)$$

Values of the temperature drop are shown in Table 8-3 as a function of the ratio d_2/d_1 . The values shown for the temperature drop as a function of watts per meter length of pipe are based on an assumed normalized value of $\sigma = 1$ W/m·°C, assumed for the thermal conductivity of typical salts. It is obvious that for a specific case where a known value of σ is available, the results shown in column 3 of Table 8-3 can be divided by the value of σ . It is apparent from this tabulation that as the thickness of the frozen section increases, the temperature difference increases significantly. As a result values are shown to a diameter ratio of 5. The volume of fused material relative to the total volume within the tube is given in column 5. It can be seen that for d_2/d_1 of 5, 96 percent of the material has changed state.

To further illustrate the characteristics of heat transfer for a salt in tube storage device, the stored energy in kilowatt hours for a 0.1 meter diameter tubular container with a typical value of heat of

Table 8-3. Temperature Drops for a Salt in Tube Storage System

(1)	(2)	(3)a	(4)	(5)	(6)b	(7)c	(8)d	(9)e
d_2/d_1	$\text{Log}(d_2/d_1)$	$\Delta T/P\ell$ °C/(W·m)	$(d_1/d_2)^2$	$\Delta v/v$	U_w kWh	ΔT salt °C	ΔT water °C	ΔT salt, $d = 0.0$ °C
1.0	0	0	1	0	0	0	0.32	0
1.1	0.04	0.015	0.83	0.17	0.32	8	0.32	1.8
1.5	0.17	0.064	0.44	0.55	1.01	32	0.32	8
2.0	0.30	0.109	0.25	0.75	1.37	55	0.32	14
3.0	0.48	0.175	0.11	0.89	1.62	88	0.32	22
5.0	0.69	0.256	0.04	0.96	1.75	128	0.32	32

a) calculated for $\sigma = 1 \text{ W/m } ^\circ\text{C}$

b) $H_f = 100 \text{ cal/gr} = 4.18 \times 10^5 \text{ watt sec/kg}$, $\delta = 2,000 \text{ kg/m}^3$

c) $P/\ell = 500 \text{ W/m}$, $d = 0.1 \text{ m}$

d) $P/\ell = 500 \text{ W/m}$, $d = 0.1 \text{ m}$, $v = 1 \text{ m/sec}$

e) $P/\ell = 125 \text{ W/m}$, $d = 0.05 \text{ m}$

fusion, H_f , of 100 cal/g is shown in column 6. If a different value of H_f is assumed, it is a simple matter to modify the values given in column 6 to represent the value of heat of fusion for the salt considered. The resulting temperature difference for a power density flow of 500 W/m of pipe length for the case of a pipe diameter of 0.1 m is shown in column 7. The resulting temperature difference becomes large when 96 percent of the available material is fused. If water cross flow at 1 m/s over the heat storage tubes is considered, the resulting temperature drop is shown in column 8.

The temperature drop which would occur with a liquid organic fluid will be 2 to 6 times greater than that shown for water. If steam is produced, the temperature drop for the 0.1 m diameter tubes will be approximately 30 to 40 times the value shown in column 8, or about 10° to 15°C, because although the film transfer coefficient is appreciably lower for steam, the velocities typically involved are 10 or more times greater than for water.

It is possible to relate the withdrawal rate to the energy storage and time to withdraw 96 percent of the energy by the relationship

$$P/l = \frac{0.96 U/t}{T_w} = \frac{\pi d_2^2 H_f \delta}{4 T_w 3600} \quad (8-8)$$

where P/l is the withdrawal rate, w/m of length,

U/t is the heat storage, Wh/m of pipe length,

T_w is the time in hours to withdraw 96% of the total energy
when a uniform withdrawal rate is employed, hr,

d_2 is the inner container diameter, m,

H_f is the heat of fusion for the material, W·s/kg,

δ is the salt density, kg/m³.

Consider again the example for a withdrawal rate of 500 W/m^2 using 0.1 m diameter tubes filled with a salt, having a heat of fusion of 100 cal/g, or $4.18 \times 10^5 \text{ W}\cdot\text{s/kg}$ (or 116 kWh/kg), and a typical salt density of $2 \times 10^3 \text{ kg/m}^3$. The resulting time to withdraw 96 percent of the thermal energy is 3.5 hours. If a withdrawal time of 3.5 hours is considered for a pipe diameter of 0.05 m, instead of 0.1 m, the power flux is reduced by the ratio of the square of the diameters, to 125 W/m^2 , and the resulting temperature drop in the salt is shown in column 9 of Table 8-3. The consequences of using a smaller pipe diameter are to also decrease the temperature drop through the fluid film outside of the pipe.

If the salt is contained within the tubular containers and an organic fluid is circulated around them, even for the case where the tubes are essentially touching, approximately 20 percent of the volume will be liquid. If a reasonable spacing is allowed between the tubes containing the salt, say 0.1 of the tubular diameter, then approximately 35 percent of the total heat storage volume must be filled with the fluid. If this is an organic fluid, a rather large amount of expensive material is needed. In view of this, it appears desirable to examine the case where tubes, in which the heat transfer or working fluid is contained, are placed within a large container and is then filled with the salt (see Figure 8-1). With this arrangement, 90 percent or more of the total volume can be occupied by the salt and only 10 percent or less is required for the transfer fluid. If a rectangular array of pipes is arranged which are spaced 4 diameters on center, the volume in the salt becomes 90 percent and that of the liquid, 10 percent of the total volume. The maximum ratio of diameters is somewhat in excess

of the 4 and probably near 5, in which case the temperature drop relation should be similar to those calculated for the preceding examples, except that it is now possible to design a heat exchanger where there is significantly less organic fluid required in the system.

When properly designed, the heat of fusion type of storage system has the potential for yielding a rather small change in output temperature and a relatively low loss of availability through the system. As determined from the preliminary analysis, it is however, necessary to use relatively small diameter pipes if the temperature drop is to be controlled to within reasonable levels while still maintaining a reasonably short withdrawal time.

Approximate cost of heat storage material can be obtained by

$$$/kWh_t = 3.6 \times 10^6 C_m / H_f \quad (8-9)$$

where C_m is the cost of the salt, \$/kg

H_f is the specific heat, Wh/kg.

If a nominal value of \$0.03/kg is assumed for the salt, and 100 cal/g (or 4.18×10^5 W·s/kg for the specific heat of the salt) the cost is about \$0.30/kWh_t. This compares very favorably with the previous examples where the cost of stored thermal energy in an organic fluid was about 50 times higher. The cost of the container, the piping within the salt material, and the cost of the heat transfer fluid in the pipes, if it becomes a significant fraction of the total volume, must be added to the cost of the basic material.

For the case where 0.05 m diameter pipes are spaced 4 diameters apart and a 1,000 m³ tank of 1/4" steel is used for the housing, approximate cost can be obtained from the relation

$$$/kWh_t = 3.6 \times 10^6 (C_m/H_f + 1.1 C_t/H_f\delta + 16 C_p/H_f\delta) \quad (8-10)$$

where C_m = cost of salt, \$0.03/kg

$$H_f = 4.18 \times 10^5 \text{ W}\cdot\text{s/kg}$$

$$\delta = 2,000 \text{ kg/m}^3$$

$$C_t = \text{tank costing rate} = \$20/\text{m}^3 \text{ in size shown}$$

$$C_p = 3 C_{pm} = 3 \times \$3/\text{m} \text{ for } 0.05 \text{ m diameter pipe, where } C_{pm} \text{ is the cost of pipe, } \$/\text{m}.$$

The result is

$$\begin{aligned} \$/kWh_t &= 0.26 + 0.09 + 0.83 \\ &\quad \underbrace{\hspace{1cm}}_{\text{salt}} \quad \underbrace{\hspace{1cm}}_{\text{tank}} \quad \underbrace{\hspace{1cm}}_{\text{pipe}} \\ &\approx \$1.20/kWh_t. \end{aligned}$$

Actual costing rates will be greater when insulation and installation are added. This is only an order of magnitude figure for mass produced units.

Since the primary objective of this section is to examine storage concepts as opposed to deriving detailed costs, no attempt will be made to achieve precise cost calculations. Before precise costs can be estimated, careful determination of the types of salt or heat storage material is required, along with an examination of the types and costs of liners which can be installed in the containers as well as coating for the pipes to prevent excessive corrosion.

9.0 CONCLUSIONS

As a result of this investigation it has been determined that parabolic trough collectors are more cost effective for use in converting solar energy to heat than through other line focus systems included in this study. The minimum cost for thermal power delivered from an optimized PTC collector field for power plant capacities in the range from 2 to 20 MW_e is \$240/kW_t at 300°C. This compares with \$350/kW_t from an RSC collector field at 250°C, \$415/kW_t from a MAC collector field at 250°C and \$1000/kW_t from a CPC collector field at 150°C. Corresponding costs for electrical energy production are 6.5¢/kWh_e from a PTC collector system, 8.5¢/kWh_e from an RSC collector system, 9.7¢/kWh_e from a MAC collector system and 31.8¢/kWh_e from a CPC collector system. Electrical energy production costs are insensitive to power plant capacities, but plant sizes of 20 to 30 MW_e appear to be near optimum.

The working fluid in all systems is steam, with water returned from the condenser to the field and steam generated in the collectors supplied to the turbine. Organic fluids were found to be technically suitable in the temperature range from 100 to 300°C, with acceptable thermodynamic efficiencies of Rankine cycle heat engines, but costs were higher than for steam turbines. Although the cost for equipment with ammonia as a working fluid is comparable to steam turbines, operating temperatures near 100°C were not found to be optimal.

An appropriate organic fluid in the 100°C working temperature region is ammonia, in the 200°C region methanol is acceptable and in the 300°C region a mixture of pyridine and water is better than Fluorinol-85. Heat engines suitable for organic fluids have been conceptually designed in this study and cost estimates are provided for the turbogenerator and

appurtenant power plant equipment. The costs are parameterized in a form suitable for inclusion in a computerized optimization program. Generally, costs for the turbogenerator above is about \$750/kW (1975 prices) for methanol and pyridine/water fluids and \$375/kW for ammonia systems which is near the cost for steam turbines. Appurtenant costs of condenser, cooling tower and plant piping raises the power plant costs considerably. When collector and storage costs are added, capital costs of solar thermal electric power plants are very high compared to conventional power plants.

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APPENDIX A

WORKSHEETS FOR ESTIMATING MATERIALS, LABOR AND OVERHEAD COSTS

- A-1 Honeywell Trough 4' x 8'
- A-2 Capital Investment Summary for Honeywell Trough
- A-3 Sandia Trough 8' x 12'
- A-4 Sandia Trough 7' x 12'
- A-5 Capital Investment Summary for Sandia Trough
- A-6 ITEX Capital Equipment and Labor
- A-7 ITEX Cost Estimate
- A-8 Capital Investment Summary for ITEX Collector
- A-9 General Atomic Rotation 40' x 25'
- A-10 General Atomic Double Size
- A-11 CPC Capital Equipment and Labor
- A-12 CPC 3' x 5'
- A-13 CPC 4' x 8'
- A-14 Capital Investment Summary for CPC Collector

A-1 HONEYWELL TROUGH 4' x 8' (Ea/Min)

CAPITAL EQUIPMENT AND LABOR

(\$000)

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL			
SHELL	6	Decoilers	12	72	6	3	36
	6	Straighteners	15	90	6	3	36
	6	Wobble Expanders	10	60	6	3	36
	6	Curing Ovens	20	120	6	3	36
	6	Wobble Cutters	10	60	6	3	36
	6	Strip Slitters	10	60	6	3	36
	6	Honeycomb Coaters	10	60	6	3	36
	2	Box Beam Roll Forms	25	50	2	3	12
	1	Edge Roll Form	20	20	1	3	6
	1	Automatic Coil Handler	60	60	1	3	6
	1200'	Conveyor Equipment	100/ft	120	-	-	-
	6	Squeeze Rolls	10	60	6	3	36
	6	Vapor Degreasers	15	90	6	3	36
	6	Wobble Section Turners	5	30	1	3	6
	6	Curing Ovens	15	90	6	3	36
	6	Shears	10	60	6	3	36
	6	Stackers	15	90	6	3	36
	6	Feeders	15	90	6	3	36
	36	Automatic Assembly Tools	10	360	36	3	216
	6	Conformation Tables	20	120	-	-	-
	6	Frame Benders	20	120	6	3	36
	6	Assembly Conveyors	20	120	84	3	504
	6	Material Handling Equipment	20	120	36	3	216
	-	Miscellaneous Equipment	-	420	40	3	120
	-	Material Handlers	-	-	20	3	120
	-	Line Foreman	-	-	10	6	120
	-	Equipment Installation @ 25%	-	686	-	-	-
	6	Gaging and Controls	-	200	18	4	144
		TOTAL SHELL LINE		<u>\$3,428</u>	<u>333</u>		<u>\$1,974</u>
REFLECTOR	6	Glass Draping Ovens	100	600	12	3	72
	120	Conveyor Handrails	3	360	6	3	36
	6	Glass Loading Devices	10	60	6	3	36
	6	Cooling Tunnels	60	360	6	3	36
	1200'	Glass Conveyors	100/ft	120	-	-	-
	6	Washing Stations	30	180	6	3	36
	12	Silver Mirror Lines, Automatic	200	1200	24	4	144
	18	Inspection Stations	20	360	18	4	144
	18	Packing Stations, Automatic	20	360	18	3	108
	-	Equipment Installation @ 25%	-	1130	-	-	-
	-	Material Handlers	-	-	10	3	60
	-	Line Foreman	-	-	10	6	120
	-	Miscellaneous Equipment	-	920	24	3	144
		TOTAL REFLECTOR LINE		<u>\$5,606</u>	<u>110</u>		<u>\$936</u>

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE EACH	COST TOTAL	NO. OPERATORS	HOURLY RATE	ANNUAL COST
SUPPORT	6	Steel Storage Racks	1	6	-	-	-
	6	Steel Plate Feeders	5	30	6	3	36
	12	Automatic Oxynacetylene Cutters	3	60	12	3	72
	12	Edge Grinders, Automatic	10	120	12	3	72
	12	Press Brakes	35	420	12	3	72
	6	Band and Cutoff Cutters	10	60	6	3	36
	6	Band Drills	40	160	6	3	36
	24	Automatic Welders	20	480	24	3	144
	6	Automatic Screw Machines	60	360	6	4	48
	6	Automatic Facing Machines	40	240	6	4	48
	12	Assembly Stations	40	480	36	3	216
	-	Miscellaneous Equipment	-	483	-	-	-
	6	Material Handling Equipment	20	120	54	3	324
	-	Material Handlers	-	-	18	3	108
	-	Line Foremen	-	-	10	6	120
	-	Equipment Installation @ 25%	-	755	-	-	-
	-	Phosphate Line, Installed	-	1,000	2	3	12
	-	Paint Line, Installed	-	1,200	24	3	144
	TOTAL SUPPORT LINE			\$5,974	140		\$1,368
DRIVES	12	Chain Parts Multi-slides	60	720	6	3	36
	6	Sprocket Cutters	35	210	6	4	48
	1	Automatic Screw Machine	60	60	1	4	8
	1	Automatic Face Grinder	40	40	1	4	8
	24	Sensor Assembly Stations	10	240	24	3	144
	24	Assembly Stations	20	480	72	3	432
	72	Automatic Assembly Tools	2	144	-	-	-
	-	Miscellaneous Equipment	-	379	-	-	-
	-	Material Handlers	-	-	10	3	60
	-	Line Foremen	-	-	5	6	60
	-	Equipment Installation @ 25%	-	568	-	-	-
	12	Inspection Stations	10	120	12	4	96
	TOTAL DRIVES LINE			\$2,961	137		\$802
ABSORBER	6	Tubing Cutters	10	60	6	3	36
	2	Glass Sealers	60	120	4	3	24
	6	Tube Welders	40	240	6	3	36
	1	Evacuation and Test Carroussel	100	160	4	3	24
	1	Plating Equipment	80	30	2	3	12
	1	Tube Feeder	20	20	-	-	-
	-	Miscellaneous Equipment	-	124	-	-	-
	-	Material Handlers	-	-	3	3	18
	-	Line Foremen	-	-	3	6	36
	-	Equipment Installation @ 50%	-	372	-	-	-
	TOTAL ABSORBER LINE			\$1,116	28		\$186

4' x 8' (2.97 m²) Projected
(Net No. = 588,438/Yr)

COMPONENT	DESCRIPTION AND SPECIFICATIONS	DIMENSIONS (INCH)	MATERIAL WEIGHT (LBS)	QUANTITY REQUIRED	(QTY) MATERIAL COST (\$)	UNIT PRICE (\$)	TOTAL COST (\$)	PERCENTAGE OF TOTAL (\$)
Shell	Al-Quoted Steel, Type 2, 0.025", 1.067 sq ft ²	2' 0" x 4' 0"	.95	84.7 ft ²	\$2,571.4	\$30.22/ft ²	\$11,152	1.14%
Shell	End Plates, Al-Alloy 4.052, 0.125", .007 sq ft ²	2' 12" x 42"	.95	14.70 #	0,000.4	0.0001/#	5.338	.05%
Shell	Anti-Paper Reinforcement 10 (14) 1/2", impregnated	40 ft ² x 1" thick	.95	3,809 ft ²	2,005 ft ²	1.309/ft ²	2,103	.45%
Shell	Aluminum Strip, Alloy 3.052, .01" x 0.32", .006 sq ft ²	83 ft x 1", 26 ft x 4"	.95	0.44 #	5,023 #	0.0001/#	2,312	.39%
Shell	Miscellaneous Hardware						2,942	.50%
Shell	Aluminum Salvage (4% of Volume)		.94	.26 #	152 #	\$0.15/#	(23)	(.02%)
Shell	Epoxy Resin		.90	1 #	600 #	0.55/#	340	.35%
Shell	Volume Discount on Shell Materials @ Average 15%						(3,903)	(.40%)
Shell	Net Shell Materials Cost						\$71,052	2.54%
								(\$72,200) ²
Reflector	Liquid Gases, 10 Panels Cut to Size	12" x 48" x 0.100"	.90	44.4 ft ²	26,155 ft ²	0.16/ft ²	4,303	.70%
Reflector	Silver Coating	5' x 8' x 5 x 10 ⁻⁵ "	.99	2063	26,155 ft ²	\$1.90/ft ²	107	.28%
Reflector	Adhesive Backing or Film Cushion	5' x 8' x .001"	.95	5.4 lbs	2,000 lbs	.10/lb	1,400	1.40%
Reflector	Net Reflector Assembly Cost						\$5,810	8.38%
								(\$13,360) ²
Support	Low Carbon Steel Plate, 0.363 sq ft ²	30" x 36" x 1/4"	.95	32.5 lbs	42,000 lbs	0.145/#	4,704	.10%
Support	Low Carbon Steel Plate, 0.282 sq ft ²	30" x 18" x 1/2"	.95	345 lbs	34,000 lbs	0.145/#	2,940	.24%
Support	Drill Stem, S. S., 0.2014 cu ft ³	2' (1 1/2" x 3/4")	.95	0.320 lbs	192 lbs	1/8	578	.48%
Support	Nuts & S.	2' (1.2" OD, 3/4" ID)	.90	0.651 lbs	385 lbs	3/4	1,156	1.41%
Support	Miscellaneous Hardware						1,173	1.50%
Support	Steel Salvage		.94		(2,200 lbs) @ .02/#		(45)	(.43%)
Support	Net Support Materials Cost						\$11,075	22.75%
								(\$24,376) ²
Drive	Alum. Steel Tube for Chain Wdg.	3/4" L, 1/4" OD, 1/8" ID	.95	67.6"	340 lbs	1.00/lb	340	.58%
Drive	Alum. Steel Tube for Chain Wdg.	3/4" L, 3/8" OD, 1" ID	.95	80.0"	432 lbs	1.00/lb	432	.70%
Drive	Alum. Steel Strip for Chain Wdg.	3/4" W, 1/16" Thick	.95	83.1"	480 lbs	.50/lb	240	.39%
Drive	Alum. Steel Rod for Chain Wdg.	1/8" Ø	.95	93.8"	152 lbs	.75/lb	114	.19%
Drive	Alum. Steel Plates for Sprockets Wdg.	4" Quad, x 1/4" Thk.	.95	3 each	1,230 ea.	.75 ea	920	1.50%
Drive	Alum. Steel Rod 1/2" Dia. for Sprockets	1" Long x 2" Span.	.95	5 each				
Drive	Alum. Steel Rod 1.5" Dia. for Sprockets	2' (1" Long x 1.5" Ø diam.)	.95	2 each	340 lbs	.75/lb	255	.49%
Drive	Alum. Steel Rod 1" Dia. for Sprockets	2" Long x 1" Wide	.95	1 each				
Drive	Alum. Steel Rings, Cut to Spec Blank	11 5/8" OD, 0.5" ID, 7 3/8" THK	.90	2 4/4 ea	1,710 lbs	.40/lb		.27%
Drive	Miscellaneous Materials						1,177	2.08%
Drive	Steel Salvage (4% of Volume)		1.00	2.4	147 ea	.05/lb	(13)	(.02%)
Drive	Plastic Mount Bearing Blocks 1.5" Bore		.90	1 1/4	750 ea	5.90 ea	2,265	3.75%
Drive	Shafts, Bolts, 2" Pipe Sch 40 3.65 lbs/ft	80" Long	.95	90"	4,700 ft	.25/lb	2,475	5.09%
Drive	Carbon steel Salvage (4% of Volume)					.024/lb	(17)	(.03%)
Drive	Double Reduction Drivegear 400 - 1 Kva's 1/4 H P 3.0 RPM, Output, 2275-30 RPM, 220/100 V 3ø	1 Ratio Motor/4 Panels 0.12 PM Output	1.00	3/4	147 ea	376 ea	70,000	119.11%
Drive	Salvage Discount on Ratio Motors 2 40% (Minimum Gear)							
Drive	Bolted to the shafts, dimensioned, less 40% volume discount		1.00	1/4	147 ea	12.00	1,764	2.80%
Drive	Hardware for Sensor Wdg.		.85	1/4	175	4.00	620	1.11%
Drive	2 Photo Diodes 2 Transistors 3 tubes envelope Miscellaneous Hardware							
Drive	Net Drive Materials Cost						\$55,233	52.71%
								(\$79,620) ²
Joiner	Carbon Steel Tube, 3/4" OD x .049" Wall	90" Long	.95	8.25	4,955 ft	.1515/ft	667	1.05%
Joiner	Nonferrous Glass Tube, 2" OD x .032"	90" Long	.90	1,695 lbs	931 lbs	.7	745	1.27%
Joiner	Metal End Guard Rings	2" OD x 3/4" ID	.95	2	1,290 ea	.50 ea	620	1.05%
Joiner	Eccentricity Expansion Bolts Rings	1 1/2" OD x 3/4" ID	.95	2	1,290 ea	.50 ea	620	1.05%
Joiner	Miscellaneous Materials 2 Hardware						2,405	3.44%
Joiner	Net Joiner Materials Cost						\$7,412	6.47%
								(\$116,000) ²
TOTAL	Net Total Materials Cost							
	Shell	\$21,052	36.41	11.30				
	Reflector	8,770	8.92	3.36		Construction & Building	5,000,000	
	Support	11,075	10.70	4.32		Machinery & Equipment	22,250,000	
	Drive	55,233	92.82	31.63				
	Joiner	7,412	6.47	2.15		Labor, Plant	5,350,000	
	TOTAL	99,542	210.91	536.46		Materials	101,033,000	
						Overhead	8,000,000	
						Wfg. Cost	\$145,023,000	
							\$195,522,000	

A-2 HONEYWELL SUMMARY

CAPITAL INVESTMENT

(\$000)

Machinery and Equipment	\$19,129
Engineering @ 20%	<u>3,826</u>
Total Machinery and Equipment	<u>\$22,955</u>
Building 150,000 Sq. Ft. @ \$23/sq. ft.	\$ 3,750
Facilities @ 20% of Equipment	4,591
Landscaping, Roads, Docks and Grounds @ 10% Building	375
Architect and Planning @ 10% Building	<u>375</u>
Total Building and Improvements	<u>\$ 9,091</u>
Land 100 acres @ \$10,000/ac	<u>1,000</u>
TOTAL CAPITAL INVESTMENT	<u>\$33,046</u>

WESTINGHOUSE # 6 STATEMENT FORMAT
(\$000)

#6 Line No.		
10	<u>Direct Cost, Labor</u> (778 Operators)	\$ 5,356
11	<u>Direct Cost, Materials</u> (\$88,438 Troughs Not)	<u>97,645</u>
	Supervision (5 supervisors @ \$20,000/yr)	100
	Tooling	950
	Power (10,000 MWH @ \$.057/KWH)	570
	Freight (\$.01/lb on Raw Materials)	1,540
	Pension and Insurance @ 15% of Direct Labor	803
	Maintenance (10% of Capital Investment/Yr in Machinery and Equipment)	1,913
12	<u>Direct Cost, Factory Expense</u>	<u>5,676</u>
17	Direct Product Cost (Sum Lines 10, 11, 12)	<u>\$108,677</u>
19	Manufacturing (5 Engineers @ \$15,000/Yr + 15%)	86
20	Engineering (2 Engineers @ \$20,000/Yr + 15%)	46
21	Marketing (2 Salesmen @ \$15,000/Yr + 15%)	34
22	Administration and General (Administration, Clerical + Security + 15%)	483
24 a	TOTAL OPERATING MANAGED COSTS	<u>649</u>
24 b	Headquarters	500
26	Depreciation @ 10%/Yr on Equipment	2,296
	5%/Yr on Building	<u>454</u>
		\$ 2,750
	Insurance and Taxes @ 2% of Capital Investment (Total)	600
	Holidays and Vacations (20 Days Average/Person)	482
27	TOTAL	<u>1,032</u>
28	Division Committed Costs (Sum Lines 26 and 27)	<u>3,832</u>
30	TOTAL OPERATING COSTS (Sum Lines 17, 24, and 28)	<u>\$115,158</u>
		(\$192.50/Trough)
		(\$64.75/m ²)
	Headquarters	<u>0.37</u>
		<u>\$15.12</u>

A-3 8' x 12' (2 ea/min)

CAPITAL EQUIPMENT AND LABOR

(\$000)

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL			
SHELL	4	Decoilers	12	24	4	3	24
	4	Straighteners	15	30	4	3	24
	6	Wipe Expanders	10	60	6	3	36
	6	Curing Ovens	20	120	6	3	36
	6	Wipe Cutters	10	60	6	3	36
	4	Strip Slitters	10	40	4	3	24
	4	Honeycomb Coasters	10	40	4	3	24
	2	Box Beam Roll Forms	25	50	2	3	12
	1	Edge Roll Form	20	20	1	3	6
	1	Automatic Coil Handler	60	60	1	3	6
	1200'	Conveyor Equipment	100/Ft	120	-	-	-
	4	Squeeze Rolls	10	40	4	3	24
	4	Vapor Degreasers	15	90	6	3	36
	4	Wipe Section Turners	5	30	1	3	6
	4	Curing Ovens	15	60	4	3	24
	4	Shears	10	40	4	3	24
	4	Stackers	25	100	4	3	24
	4	Feeders	25	100	4	3	24
	36	Automatic Assembly Tools	10	360	36	3	216
	4	Conformation Tables	20	120	-	-	-
	4	Frame Benders	30	120	8	3	48
	2	Assembly Conveyors	60	120	24	3	504
	4	Mail Handling Equipment	20	120	56	3	216
	-	Miscellaneous Equipment	-	252	57	3	611
	-	Material Handlers	-	-	20	3	120
	-	Line Foremen	-	-	10	6	120
	-	Equipment Installation @ 25%	-	582	-	-	-
	4	Gaging and Controls	-	150	-	4	96
		TOTAL SHELL LINE		<u>\$2,908</u>	<u>266</u>		<u>\$1,821</u>
REFLECTOR	4	Glass Draping Ovens	150	600	8	3	48
	120	Conveyor Mandrels	3	360	6	3	36
	4	Glass Loading Devices	15	60	4	3	24
	4	Cooling Tunnels	90	360	4	3	24
	1200'	Glass Conveyors	100/Ft	120	-	-	-
	4	Washing Stations	45	180	4	3	24
	12	Silver Mirror Lines, Automatic	100	1,200	24	3	144
	18	Inspection Stations	20	360	18	4	144
	18	Packing Stations, Automatic	20	360	18	3	108
	-	Equipment, Installation	-	1,130	-	-	-
	-	Material Handlers	-	-	10	3	60
	-	Line Foremen	-	-	10	6	120
	-	Miscellaneous Equipment @ 20%	-	920	21	3	126
		TOTAL REFLECTOR LINE		<u>\$5,650</u>	<u>127</u>		<u>\$853</u>

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL			
SUPPORT	4	Steel Storage Racks	1	4	-	-	-
	4	Steel Plate Feeders	5	20	4	3	24
	8	Automatic Oxyacetylene Cutters	5	40	8	3	48
	8	Edge Grinders, Automatic	10	80	8	3	48
	8	Press Brakes	40	320	8	3	48
	4	Base and Gusser Cutters	10	40	4	3	24
	4	Base Drills	40	160	4	3	24
	16	Automatic Welders	20	320	16	3	96
	4	Automatic Screw Machines	60	240	4	4	32
	4	Automatic Facing Machines	40	160	4	4	32
	8	Assembly Stations	50	400	32	3	192
	-	Miscellaneous Equipment	-	357	-	-	-
	4	Material Handling Equipment	20	80	36	3	216
	-	Material Handlers	-	-	14	3	108
	-	Line Foreman	-	-	10	6	120
	-	Equipment Installation @ 25%	-	555	-	-	-
	1	Phosphate Line, Installed	-	1,000	2	3	12
	1	Paint Line, Installed	-	1,200	24	3	144
		TOTAL SUPPORT LINE		<u>\$4,976</u>	<u>122</u>		<u>\$1,268</u>
DRIVES	12	Chain Ports Multi-Slides	60	720	6	3	36
	2	Sprocket Cutters	60	120	2	4	16
	1	Automatic Screw Machine	60	60	1	4	8
	1	Automatic Face Grinder	40	40	1	4	8
	8	Sensor Assembly Stations	12	96	8	3	48
	8	Assembly Stations	20	160	48	3	288
	48	Automatic Assembly/Tools	2	96	-	-	-
	-	Miscellaneous Equipment	-	258	-	-	-
	-	Material Handlers	-	-	7	3	42
	-	Line Foreman	-	-	3	6	36
	-	Equipment Installation @ 25%	-	384	-	-	-
	4	Inspection Stations	20	80	3	4	64
		TOTAL DRIVES LINE		<u>\$2,018</u>	<u>84</u>		<u>\$596</u>

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE	COST	NO.	HOURLY	ANNUAL
			EACH	TOTAL		RATE	COST
ABSORBER	4	Tubing Cutters	12	48	4	3	24
	1	Class Sailer	60	60	2	3	12
	4	Tube Welders	40	160	4	3	24
	1	Evacuation and Test Carroussel	80	80	3	3	18
	1	Plating Equipment	80	80	2	3	12
	1	Tube Feeder	30	30	-	-	-
	-	Miscellaneous Equipment	-	92	-	-	-
	-	Material Handlers	-	-	3	3	18
	-	Line Foremen	-	-	3	6	36
	-	Equipment Installation @ 50%	-	225	-	-	-
TOTAL ABSORBER LINE				\$925	21		\$144

A-4 SANDIA TROUGH

$$7' \times 12' = 7.8/m^2$$

244,072 Panels/Yr

Project Area: 8' x 12'
Actual Area: 9.25' x 12'

			Per Panel	$\$/m^2$
SHELL	1	Al Steel .017" 60" x 148" x 4 (\$8.2005/lb)	58.25	6.55
	2	End Plate Materials (\$0.665/lb)	28.25	3.20
	3	Honeycomb 2" x 9' x 12'	23.16	2.64
	4	Al Scrap \$0.607/lb	23.58	2.64
	5	Miscellaneous Hardware	10.00	1.12
	6	Al Salvage \$0.15/lb	(.12)	(.01)
	7	Epoxy \$0.55/lb	1.74	.20
	8	Volume Discount @ 15%	(21.94)	(2.46)
		TOTAL SHELL	\$124.32	\$13.93/m ²
REFLECTOR	9	Floor Glass 27 ea 12" x 48"	21.60	2.42
	10	Silver Backing	3.42	.38
	11	Adhesive	1.25	.14
		TOTAL REFLECTOR	\$26.28	\$2.94
SUPPORT		Steel Plate 1/4" x 60" x 24"	29.65	3.32
		Steel Plate 1/2" x 12" x 24"	11.84	1.33
		Stem 1" diam	1.77	.20
		Rollers 3" ID, 2" OD	3.20	.36
		Miscellaneous	4.00	.45
		Salvage Steel	(.58)	(.06)
		TOTAL SUPPORT	\$49.88	\$5.60
DRIVE	12	Chain Materials \$1.00/lb	2.13	.24
	13	Sprocket Blanks \$0.5/lb	1.58	.18
	14	Shafting \$0.75/lb	.98	.10
	15	Ring Blanks \$0.3/lb	5.04	.56
	16	Miscellaneous Hardware	4.00	.45
	17	Steel Salvage \$0.05/lb	(.12)	(.01)
	18	Bearing Blocks \$0.2/lb	7.00	.78
	19	Gear Motor Net Per Panel (1) \$1/lb	285.96	32.05
		Switch, Solid State, Net Per Panel (1)	12.00	1.34
		Sensor Materials, One Per Panel	4.72	.53
		TOTAL DRIVE	\$323.29	\$36.24/m ²
ABSORBER		Carbon Steel Tube 1" x .056 x 12'	2.28	.26
		Glass Tube 4" x .062 x 12'	7.40	.83
		Seals	2.00	.22
		Bellevue	2.00	.22
		Miscellaneous	4.00	.45
		TOTAL ABSORBER	\$17.68	\$1.98/m ²
		TOTAL MATERIALS	\$541.47	\$60.70/m ²
		Labor and Overhead @ 20%	108.29	12.11
		TOTAL MANUFACTURING COST	\$649.76	\$72.81
		Shipping and Field Installation @ 25% of Materials	135.36	15.17
		Feeds, Installed	52.00	5.85
		Seal Joints, Installed	25.00	2.80
		TOTAL	\$862.12	\$96.14/m ²

A-5 CAPITAL INVESTMENT

(\$000)

Machinery and Equipment	116,377
Engineering @ 20%	3,275
TOTAL MACHINERY AND EQUIPMENT	<u>119,652</u>
Building 150,000 Sq. Ft @ \$25/Ct ²	3,750
Facilities @ 20% of Equipment	3,930
Landscaping, Roads, Docks and Grounds @ 10% Building	375
Architect and Planning @ 10% Building	375
TOTAL BUILDING AND IMPROVEMENTS	<u>8,430</u>
Land, 100 Acres @ \$10,000/ac	<u>1,000</u>
TOTAL CAPITAL INVESTMENT	<u>\$29,082</u>

SANDIA # 6 STATEMENT FORMAT

6 Line No.

(\$000)

10	Direct Cost, Labor (722 Operators)	11,637
11	Direct Cost, Materials (190,146 Troughs Yot)	101,313
	Supervision (5 Supervisors @ \$20,000/Yr)	100
	Tooling	950
	Power (10,000 kWh @ \$ 037/kWh)	370
	Freight (\$.01/lb on Raw Materials)	1,540
	Pension and Insurance @ 15% of Direct Labor	696
	Maintenance (10% of Capital Investment/Yr in Machinery and Equipment)	1,638
12	Direct Cost, Factory Expense	5,394
17	Direct Product Cost (Sum Lines 10, 11 and 12)	111,244
19	Manufacturing (5 Engineers @ \$15,000/Yr + 15%)	26
20	Engineering (2 Engineers @ \$20,000/Yr + 15%)	46
21	Marketing (2 Salesmen @ \$15,000)	34
22	Administration and General (Administration, Clerical + Security + 15%)	483
24 a	TOTAL OPERATING UNMAGED COSTS	639
24 b	Corporate Expenses (Same as above)	639
26	Depreciation @ 10%/Yr on Equipment 5%/Yr on Building	1,065 422
		<u>15,387</u>
	Insurance and Taxes @ 3% of Capital Investment (Total)	382
	Holidays and Vacations (20 Days Average/Person)	424
27	TOTAL	<u>11,096</u>
28	Division Connected Costs (Sum Lines 26 and 27)	3,394
30	TOTAL OPERATING COSTS (Sum Lines 17, 24, and 28)	<u>\$115,286</u>
		\$387.76/Ftough (\$65.89/m ²)
	Headquarters	0.37
		<u>\$66.26</u>

A-6 ITEX
CAPITAL EQUIPMENT AND LABOR
(\$000)

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			UNIT	TOTAL			
SLATS	7	Decoilers	5	35	3	3	18
	7	Straighteners	5	35	3	3	18
	4	Hole Expanders	10	40	4	3	24
	4	Curing Ovens	20	80	4	3	24
	4	Hole Cutters	10	80	4	3	24
	4	Slitters	20	80	4	3	24
	4	Honeycomb Costers	10	40	4	3	24
	4	Box Beam Roll Forms	25	75	3	3	18
	3	Edge Roll Forms	25	75	3	3	18
	1	Coil Handler, Automatic	60	60	1	3	6
	1200'	Conveyor Equipment	100/Et	120	-	-	-
	7	Squeeze Rolls	10	70	1	3	6
	7	Vapor Degreasers	15	105	7	3	42
	7	Hole Section Turners	5	35	1	3	6
	7	Curing Ovens	15	105	7	3	42
	7	Shears	10	70	7	3	42
	7	Stackers	10	70	7	3	42
	7	Feeders	10	70	7	3	42
	36	Automatic Assembly Tools	10	350	33	3	210
	7	Contour Routers	10	70	7	3	42
	7	Frame Benders	10	70	7	3	42
	7	Slat Assembly Lines	10	70	84	3	504
	7	Material Handling Equipment	20	140	7	3	42
	-	Miscellaneous Equipment	-	420	40	3	240
	-	Material Handlers, etc.	-	-	20	3	120
	-	Line Foreman	-	-	7	6	84
	-	Equipment Installation @ 25%	-	618	-	-	-
	7	Gaging and Controls	-	200	14	4	112
		TOTAL SLATS LINE		<u>53,156</u>	<u>291</u>		<u>\$1,816</u>

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL			
REFLECTOR	3	Glass Draping Ovens	300	600	4	3	24
	120	Contour Mandrels	3	360	7	5	42
	2	Glass Loading Devices	30	60	4	3	24
	2	Cooling Tunnels	180	360	4	3	24
	1200'	Glass Conveyors	100/Ft	120	-	-	-
	4	Washing Stations	45	180	4	3	24
	24	Silver Mirror Lines, Automatic	100	1400	14	3	84
	14	Inspection Stations	20	280	14	4	112
	14	Packing Stations, Automatic	20	280	14	3	84
	-	Equipment, Installation @ 25%	-	910	-	-	-
	-	Material Handlers	-	-	10	3	60
	-	Line Foremen	-	-	10	6	120
	-	Miscellaneous Equipment @ 20%	-	728	21	3	126
		TOTAL REFLECTOR LINE		<u>\$5,278</u>	<u>106</u>		<u>\$724</u>
SUPPORT	1	Automatic Metal Saw	20	20	1	3	6
	1	Beresler	25	25	1	3	6
	1	Straightener	25	25	1	3	6
	1	Sitter	50	50	1	3	6
	1	Shear	25	25	1	4	8
	1	Automatic Bar Cutter	20	20	1	3	6
	1	Automatic Rod Cutter	20	20	1	3	6
	2	Oxyacetylene Cutters	5	10	2	3	12
	4	Automatic Welders	20	80	4	3	24
	8	Automatic Drills	20	160	8	3	48
	8	Automatic Grinders	20	160	8	3	48
	8	Assembly Stations	50	400	32	3	192
	-	Miscellaneous Equipment	-	350	-	-	-
	4	Material Handling Equipment	20	80	36	3	216
	-	Material Handlers	-	-	16	3	96
	-	Line Foremen	-	-	10	6	120
	-	Hot Dip Galvanize	500	500	4	6	48
	-	Equipment Installation @ 25%	-	481	-	-	-
		TOTAL SUPPORT LINE		<u>\$2,406</u>	<u>127</u>		<u>\$948</u>

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EXCH	TOTAL			
DRIVES	8	Sensor Assembly Stations	12	96	8	3	48
	8	Inspection and Test Station	50	400	8	4	64
	48	Automatic Assembly Tools	2	96	-	-	-
	-	Miscellaneous Equipment	-	150	-	-	-
	-	Material Handlers	-	-	4	3	24
	-	Line Foreman	-	-	4	6	48
	-	Equipment Installation @ 25%	-	185	-	-	-
		TOTAL DRIVES LINE		<u>\$927</u>	<u>24</u>		<u>\$184</u>
ABSORBER	4	Tubing Cutters	15	60	4	3	24
	4	Glass Sealers	50	100	4	3	24
	4	Tube Welders	40	160	4	3	24
	4	Evacuation and Testing Beds	100	400	8	3	48
	1	Plating Equipment	80	80	2	3	12
	1	Tube Feeder	60	60	-	-	-
	-	Miscellaneous Equipment	-	100	-	-	-
	-	Material Handlers	-	-	6	3	36
	-	Line Foreman	-	-	3	6	36
	-	Equipment Installation @ 50%	-	480	-	-	-
		TOTAL ABSORBER LINE		<u>\$1,440</u>	<u>32</u>		<u>\$204</u>

18,010,000 ft^3/yr Net
 1.7% $\times 10^8 \text{ m}^3/\text{yr}$ Net
 Mo = 101,000 Net (180,000 Gross)

[illegible]

A-8 ITEK SUMMARY
CAPITAL INVESTMENT
(\$000)

Machinery and Equipment	\$13,207
Engineering @ 20%	2,641
TOTAL MACHINERY AND EQUIPMENT	\$15,848
Building 150,000 Sq. Ft. @ \$25/ft ²	3,750
Facilities @ 20% of Equipment	3,170
Landscaping, Roads, Docks, and Grounds @ 10% of Building	375
Architect and Planning @ 10% of Building	375
TOTAL BUILDINGS AND IMPROVEMENTS	\$7,670
Land, 100 Acres @ \$10,000/ac	\$1,000
TOTAL CAPITAL INVESTMENT	\$24,518

ITEK # 6 STATEMENT FORMAT

#6 Line No.		(\$000)
10	<u>Direct Cost, Labor</u> (579 Operators)	\$ 3,776
11	<u>Direct Cost, Materials</u> (61,400 Units Net)	65,641
	Supervision (5 Supervisors @ \$20,000/Yr)	100
	Tooling	950
	Power (10,000 kWh @ \$.037/kWh)	370
	Freight (\$.01/lb Raw Materials)	1,469
	Pension and Insurance @ 15% of Direct Labor	566
	Maintenance (10% of Capital Investment/Yr)	1,321
12	<u>Direct Cost, Factory Expense</u>	4,776
17	<u>Direct Product Cost</u> (Sum Lines 10, 11, and 12)	74,193
19	Manufacturing (5 Engineers @ \$15,000/Yr + 15%)	86
20	Engineering (2 Engineers @ \$20,000/Yr + 15%)	46
21	Marketing (2 Salesmen @ \$15,000/Yr + 15%)	34
22	Administration and General (Administration, Clerical + Security + 15%)	183
24 a	TOTAL OF OPERATING MANAGED COSTS	649
24 b	Headquarters	649
26	Depreciation @ 10%/Yr on Equipment	1,585
	5%/Yr on Building	422
		2,007
	Insurance and Taxes @ 2% of Capital Investment (Total)	490
	Holidays and Vacations (20 Days Average/Person)	330
27	TOTAL	820
28	Division Committed Costs (Sum Lines 26 and 27)	2,827
30	TOTAL OPERATING COSTS (Sum Lines 17, 24 and 28)	77,649
		(\$481.22/Unit)
		\$ 44.39/m ²
		.37
		\$ 44.76/m ²

A-9 GENERAL ATOMIC ROTATOR 40' x 25'

GROSS RATE = 11 PER HOUR

= 5 1/2 MINS/UNIT

MANUFACTURER OF SUPPORT TOWERS, DRIVES AND ABSORBERS

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST (\$000)
			EACH	TOTAL (\$000)			
SUPPORT TMS	1	Semiautomatic Angle Cutter (L)	2	2	1	3	6
	2	Semiautomatic Angle Cutter (Brace)	2	4	2	3	12
	1	Semiautomatic Angle Cutter (Horizontal)	2	2	1	3	6
	4	Grinders, Semiautomatic	2	8	4	3	24
	1	Heavy Plate Cutter	5	5	2	3	12
	1	Light Plate Cutter	1	2	1	3	6
	8	Automatic Welder	10	80	8	3	48
	1	Heavy Plate Drill Stand	10	10	1	4	8
	1	Light Plate Drill Stand	5	5	1	4	8
		Miscellaneous Equipment	50	50	-	-	-
	10	Assembly Stations	10	100	60	3	360
	10	Light Cranes	20	200	5	4	40
		Material Handlers	-	-	7	3	42
		Paint Line, Complete	1,000	1,000	15	3	90
		Line Foremen	-	-	2	6	36
		Equipment Installation @ 25%		367	-	-	-
		TOTAL SUPPORTS		\$1,835	110		\$698
DRIVES	1	Pipe Cutter	5	5	1	4	8
	1	Automatic Screw Mach., H.O.	120	120	1	6	12
	1	Broach Mach.	60	60	1	6	12
	1	Grinder	10	10	1	4	8
		Miscellaneous Equipment	20	20	-	-	-
	5	Assembly Stations	5	75	25	3	150
		Material Handling Equipment	-	25	-	-	-
		Material Handlers	-	-	10	3	60
		Line Foremen	-	-	2	6	24
	5	Test Stations	10	50	5	4	40
		Equipment Installation @ 25%	-	79	-	-	-
		TOTAL DRIVES		\$394	46		\$314
ABSORBER	11	Glass Pipe Handlers	20	220	22	3	132
	1	Inspection Station	10	10	1	4	8
	1	Transfer Table	100	100	-	-	-
	1	Cut/Trim Station	10	10	1	3	6
	5	Seal Station	25	125	5	3	30
	5	Pipe Cutters	5	25	5	3	30
	10	Molding Station	10	100	10	3	60
	1	End Cutter, Automatic	5	5	1	3	6
	5	Bellows Welders	10	50	5	3	30
	11	Glass Sealers	10	110	11	3	66
	1	Plating Equipment	80	80	2	3	12

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL (\$000)			
	1	Evacuation and Test	80	80	3	3	18
		Miscellaneous Equipment	-	50	-	-	-
		Material Handlers	-	-	3	3	18
		Line Foremen	-	-	3	6	36
	11	Inspect Stations	10	110	11	4	88
	11	Assembly Stations	10	110	11	3	66
		Equipment Installation @ 25%	-	296	-	-	-
		TOTAL ABSORBER		\$1,481	94		\$606

GENERAL ATOMIC TOTALS

	CAPITAL (\$000)	NO. OPERATORS	LABOR ANNUAL COST (\$000)
SHELL	-	-	-
REFLECTOR	-	-	-
SUPPORTS	1,835	110	\$ 698
DRIVES	394	46	314
ABSORBER	1,481	94	606
TOTALS	\$3,710	250	\$1,618

A-10 GENERAL ATOMIC DOUBLE SIZE

Net/yr = 9,415 = $1.75 \times 10^6 \text{ m}^2/\text{yr}$

Projected Area = 80' wide x 25' long
= $2 \times 10^3 \text{ ft} = 185.9 \text{ m}^2$

Component

Shell Double area doubles material cost

Reflector Double area doubles material cost

Supports For double width, focal length doubles per $x^2 + y^2 = (d + x)^2$ where θ is the same

		<u>\$/Panel</u>	<u>\$/m²</u>	
Angle - 2x2x19'	2x5084 lbs	\$ 81.04	\$.44	width
All other angles	4x9186 lbs	292.72	1.58	double height
Plate - 9"x1/2"x20'	4x6381 lbs	203.28	1.09	double width
All other plates	2x10,022	256.74	1.38	height
Bar	2x2636 lbs	41.96	.22	
Pipe (arms)	2x9735 lbs	155.00	.83	width
Bearing	2x\$15 ea	66.60	.36	
Pipe (pivot)	2x2092 lbs	33.36	.18	
Lead counter WTS	2x19,413	391.72	2.11	
Steel salvage	(6409 lbs)	(34.04)	(.18)	
TOTAL SUPPORTS	192,649 lbs	\$1488.46	\$ 8.01	

Drive

Gear motor, 2x torque = 4519 in lbs

200, 1HP, 8.8 rpm, 4885 in lbs,

.68 HP out, 230/460 v, 3φ,

HW \$103.60

FMB 338 380.50

$\$484.10 \times .60 = \$290.46 \times 2 =$ \$ 580.92 \$ 3.12

Sprocket set 13.7:1 \$50 ea x 2 100.00 .53

Chain roller 2" x 15' \$30/ft 450.00 2.42

Rotating joints 35.00 .19

TOTAL DRIVES \$1165.92 \$ 6.26

Absorber

Glass tube 20" x .292 wall x 25.28' long \$3298.00 \$17.74

Metal to glass seals @ \$30 ea 60.00 .32

Pipe 12" 672.00 3.61

Pipe 8" 254.00 1.37

Welding flanges 12" 540.00 2.90

Elbows, reducer 120.00 .65

Welding flanges 8" 276.00 1.48

Ballows 90.00 .48

Misc. Hardware 33.00 .18

TOTAL ABSORBER \$5343.00 \$28.25

Single

Double

Shell

Reflector

Supports

Drive

Absorber

TOTAL - less shell & reflector

/
-
-
\$ 6.80/m²
8.90
9.90

-
-
\$ 8.01
6.26
28.25

\$ 25.60

\$42.52

GENERAL ATOMIC ROTATOR

Net Wt/Yr = 12,325 ± 1.75 × 10⁶ g²/yr²
 Effective Length = 470,725 ft/yr

Projected Area = 48' wide × 25' long × 1000 ft² = 92,000
 (Gross Production Rate = 5.75 g/m²/year)

COMPONENT	SPECIFICATIONS AND DIMENSIONS	MATERIAL UTILITY	QTY/ PAIRS	ANNUAL COST (USD)	MATERIAL UNIT COST	ANNUAL QTY (POUNDS)	ANNUAL COST (USD)	\$/g ²
SHELL	• See Construction Details							
PERFORATOR	• See Fabric and Finishing Details							
SUPPORTS	Angle, 2" x 2" x 1/4" x 10'	.00	34.4'	5,044 lbs	\$.15/lb	1763	\$40.32	\$.43
	Angle, 2" x 2" x 1/4" x 10'	.00	39.5'	2,000 lbs	.15/lb	301	15.05	.17
	Angle, 2" x 2" x 1/4" x 50'	.00	86.9'	3,345 lbs	.15/lb	103	42.05	.46
	Angle, 2" x 2" x 1/4" x 50'	.00	32.2'	1,340 lbs	.15/lb	204	10.93	.12
	Angle, 2" x 2" x 1/4" x 42"	.00	25.5'	450 lbs	.15/lb	66	5.91	.06
	Angle, 2" x 2" x 1/4" x 72"	.00	13.5'	25 lbs	.15/lb	4	.21	.00
	Plate, 42" x 1/2" x 72"	.00	246 lbs	6,883 lbs	.15/lb	1,402	\$1.81	.09
	Plate, 30" x 1/2" x 30"	.00	144 lbs	2,676 lbs	.15/lb	492	21.22	.23
	Plate, 16" x 1/2" x 18"	.00	102 lbs	1,925 lbs	.15/lb	219	15.23	.16
	Plate, 6" x 1/2" x 18"	.00	71 lbs	1,318 lbs	.15/lb	261	10.64	.11
	Plate, 6" x 1/2" x 36"	.00	87 lbs	979 lbs	.15/lb	132	.69	.01
	Bar, 3/4" x 3/4" x 9"	.00	140 lbs	2,626 lbs	.15/lb	385	20.95	.22
	Pipe, 4" Sch 40 x 31'	.00	517 lbs	9,245 lbs	.15/lb	1,406	\$7.81	.83
	Plate, 3" x 1/2" x 20"	.00	350 lbs	6,361 lbs	.15/lb	957	\$5.02	.58
	Br. TPE. 4x10 x 9" x 1/4"	.01	2	47 ea	\$15.00 ea	627	\$3.20	.34
	Plate, 2" x 1/2" x 10"	.00	110 lbs	5,201 lbs	.15/lb	480	25.49	.27
	Bar, 4" Sch 40 x 3'	.00	111 lbs	2,092 lbs	.15/lb	554	10.68	.10
	Steel Tower Height	.00	1,451 lbs	19,613 lbs	.10/lb	3,086	\$95.84	2.11
	Steel Support = 4% of Tower			(1,969 lbs)	.05/lb	293	1.023	.10
	TOTAL SUPPORT			89,567 lbs		\$12,025	\$622.31	\$6.50
				(3,407 lbs ea)				
DRIVE	Generator, 2515 in Hrs., 120000	1.0	2	38 ea	\$226 ea	5 8,100	1481.90	\$4.34
	Engine, 100 x 85 x	1.0	2	28 ea	25 ea	71	\$4.00	.51
	Chain, Fuller 1" x 10'	1.0	2	28 ea	\$ 15/ea	5.00	\$90.00	3.23
	1/2" RACE, Robot, Drive Materials	1.0	2	19 ea	6 10 ea	184	25.02	.27
	TOTAL DRIVE					115,240	\$620.92	\$6.42
ASSEMBLY	Glass Plate, 10" x 10" x .150 wall	.00	733 lbs	1,340 lbs	\$.75/lb	10,250	\$49.05	\$.91
	25.18" Gwy. Dimensions	.00	2 ea	36 ea	\$10.00 ea	176	\$6.06	.23
	Plate, 6" Sch 40, 25.3"	.00	747 lbs	14,059 lbs	.15/lb	2,189	\$12.00	\$.21
	Plate, 4" Sch 40, 25.3"	.00	261 lbs	3,314 lbs	.15/lb	397	42.33	.46
	Welding Flange, 3000-4"	1.0	216 lbs	43 ea	\$45.00 ea	1,605	\$9.04	.97
	Elbow, Reduc., 6"-4"	.00	60 lbs	61 ea	\$18.00 ea	378	20.70	.22
	Welding Flange, 6000-4"	.00	106 lb	42 ea	\$22.00 ea	560	\$9.04	.99
	Bottom	.00	3 ea	19 ea	\$30.00 ea	145	\$6.00	.62
	Miscellaneous Hardware	-	210,00 ea	189 ea	-	189	\$7.00	.11
	TOTAL ASSEMBLY					\$12,523	\$620.01	\$6.50

A-11 CPC
CAPITAL EQUIPMENT AND LABOR
(\$000)

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL			
SHELL	6	Decoilers	10	60	4	3	24
	3	Straighteners	10	30	3	3	12
	3	Slitters	20	60	3	3	18
	1	Automatic Coil Handler	60	60	1	3	6
	1	Roll Form	75	75	2	3	12
	2	Automatic Transfer Press	75	150	2	3	12
	12	Benders	15	180	12	3	72
	12	Automatic Foam Dispensers, 50 lbs/min	50	600	12	3	72
	12	Carroussel Foam Lines	100	1200	12	3	72
	1200'	Conveyor Equipment	100/ft	120	-	-	-
	1	Shears	20	40	1	3	6
	2	Stackers	10	20	2	3	12
	12	Assembly Lines	20	240	24	3	144
	12	Seam Welders, Automatic	25	300	12	3	72
	12	Material Handling Equipment	20	240	12	3	72
	-	Miscellaneous Equipment	-	420	40	3	240
	-	Material Handlers, Etc.	-	-	20	3	120
	-	Line Foremen	-	-	12	6	144
	-	Equipment Installation @ 25%	-	949	-	-	-
		TOTAL SHELL LINE		\$4,742	174		\$1,110
REFLECTOR	3	Double Decoilers	20	60	3	3	18
	3	Straighteners	10	30	3	3	18
	3	Slitters	40	120	3	3	18
	1	Automatic Coil Handler	60	60	1	3	6
	6	Roll Forms	50	300	6	3	36
	6	Stamping Presses	50	300	6	3	36
	1	Deburring Tumbler	10	10	1	3	6
	4	Gasket Extruder	60	240	4	3	24
	4	Take-up Stands	10	40	4	3	24
	4	Reflector Clip Extruders	60	240	4	3	24
	1	Bright-Dip Anodizing Facility, Continuous Strap	500	500	3	3	18
	6	Buff and Polish Stations	10	60	6	3	36
	6	Automatic Assembly Machines	50	300	6	3	36
	3	Cover Assembly Lines	-	-	3	3	18
	3	Glass Feed Lines	10	30	-	-	-
	6	Shears	20	120	6	3	36
	-	Miscellaneous Equipment	100	100	-	-	-
	-	Material Handlers, Etc.	-	-	10	3	60
	-	Line Foremen	-	-	6	6	72
	-	Equipment Installation @ 25%	-	376	-	-	-
	12	Test and Inspection Stations	10	120	12	4	96
		TOTAL REFLECTOR		\$3,006	87		\$532

ITEM	QUANTITY	EQUIPMENT OR OPERATORS	ESTIMATE COST		NO. OPERATORS	HOURLY RATE	ANNUAL COST
			EACH	TOTAL			
SUPPORT	4	Channel Receiving Rack	5	20	-	-	-
	2	Channel Feeder	20	40	2	5	12
	24	Automatic Cutting Stations	5	120	24	3	144
	24	Automatic Welders	20	480	24	3	144
	1	Automatic, Chromate Dip Line Installed	1,000	1,000	12	3	72
	1	Chromate Curing Oven, Etc.					
	1200'	Conveyor Line	100	120	-	-	-
	1	Automatic Electrostatic Spray Paint, Installed	1,200	1,200	12	3	72
	1	Paint Curing Oven, Etc.					
	12	Edge Grinders, Automatic	10	120	12	3	72
	2	Oxycetylene Plate Cutters, Automatic	10	20	2	3	12
	-	Miscellaneous Equipment	50	50	-	-	-
	4	Material Handling Equipment	20	80	56	3	216
	-	Material Handlers	-	-	18	3	108
	-	Line Foremen	-	-	8	6	96
	-	Equipment Installation @ 25%	-	262	-	-	-
TOTAL SUPPORT LINE				\$1,510	150		\$948

DRIVES

ABSORBER

NONE

12	Tube Receiver, Automatic Feed	10	120	6	3	36
12	Automatic Tube Cutters	5	60	12	3	72
12	Automatic Deburring Machines	5	60	12	3	72
12	Automatic Welders	10	120	12	3	72
12	Assembly Lines	10	120	48	3	288
-	Material Handlers	-	-	12	3	72
-	Line Foremen	-	-	6	6	72
-	Equipment Installation @ 25%	-	120	-	-	-
TOTAL ABSORBER			\$600	108		\$684

TOTAL CPC (\$000)

	CAPITAL INVESTMENT	MATERIALS	LABOR	NO. OPERATORS
SHELL	\$4,744	\$25,319	\$1,110	174
REFLECTOR	3,006	28,092	582	87
SUPPORT	3,510	11,911	948	150
DRIVES	0	0	0	0
ABSORBER	600	16,588	684	108
TOTALS	\$11,860	\$81,910	\$3,324	519

A-12 CPC 3' x 5'

$$\text{Net} = 1.75 \times 10^6 \text{ ft}^2/\text{yr}$$

$$\text{Panel} = 3' \times 5' = 15 \text{ ft}^2 = 1.25 \text{ ft}^2/\text{panel projected}$$

$$\text{Net} = 1,255,500/\text{yr}$$

$$\text{Gross} = 1,476,918/\text{yr}$$

$$= 6,988/\text{day shift} + 12 \text{ panels/day gross}$$

$$= 738/\text{hr on 1 shift}$$

COMPONENT	DESCRIPTION OR SPECIFICATION	MATERIAL UTILITY	QTY PER PANEL	ANNUAL QTY (1000)	MATERIALS UNIT COST	ANNUAL COST (\$1000)	\$/PANEL	\$/ft ²
SHELL	Frame- Al-Coated Steel, Type 2, 0.27" x 18" x 217"	.90	20.14 ft ²	20,901 lbs	0.2700/lb	5,748	6 7.00	\$ 5.03
	Bottom Rib- Al-Coated Steel, Type 2, .027" x 32" x 72"	.90	10.33 ft ²	24,328 lbs	0.2200/lb	5,352	6.70	4.06
	Inner Pan- Al-Coated Steel, Type 2, .027" x 31" x 67"	.90	10.42 ft ²	19,115 lbs	0.2200/lb	4,210	5.35	2.41
	Side Supports- Die cast aluminum, 12" x 5" x 1/4"	.90	21 ea	2,583 lbs	0.607/lb	Estimated	Estimated	Estimated
	Polyurethane foam 1.8 pc/ft x = 0.10	.90	10 ft ³	22,507 lbs	0.45/lb	10,128	8.10	5.81
	Volume Discount on Materials @ 15%	-	-	-	-	(4,283)	(5.41)	(2.44)
	Miscellaneous Hardware	-	-	-	-	1,255	1.00	0.72
	TOTAL SHELL			109,039 lbs		25,519	\$20.32	\$14.48
REFLECTION	Reflector #12 Al Coil, 6" x .032" x 50"	.90	31.05 ft ²	29,305 lbs	0.797/lb	23,425	18.46	13.39
	@ 0.4510 lbs/ft ²							
	Bracket- Al-Coated Steel, Type 2, .032" x 6" x 26.625"	.90	20 ea	5,852 lbs	0.2110/lb	1,236	0.98	0.70
	Clear Glass- Toughie Strength Flood Glass, .12" x .13" Thk, 31.5 x .125 x 67"	.90	10.3 ft ²	22,351 lbs	0.25/ft ²	5,588	4.57	3.28
	Cover Gasket- Extruded Flexible PVC	.90	8 lbs	11,299 lbs	0.23/lb	2,597	2.24	1.63
	Aluminum Solings @ 4% of Total Volume	-	-	(1,176) lbs	0.15/lb	(176)	(1.24)	(.87)
	Volume Discount @ 15%	-	-	-	-	(4,357)	(3.88)	(2.83)
	TOTAL REFLECTION			82,680 lbs		\$28,082	\$22.36	\$16.05
SUPPORT	Structural Steel Channel for Support Racks (Dimensions) 3" x 1.5" x 1/4" (16 panels/rack)	.90	64.3 lbs	80,720 lbs	8.127/lb	10,252	8 8.16	\$ 5.86
	Pads, Steel 6" x 6" x 1/2"	.90	5.18 lbs	3,092 lbs	.102/lb	307	.45	.30
	Miscellaneous Hardware	-	-	-	-	1,255	1.00	.72
	Steel Solings @ 4% of Volume	-	(2.10 lbs)	(3,286)	.05/lb	(169)	(1.22)	(.86)
	TOTAL SUPPORT			81,324 lbs		11,005	\$ 9.49	\$ 6.81
DRIVES	None	-	-	-	-	-	-	-
ABSORBER	Carbon Steel Tube, 1" OD x .056 Wall, 15,000 gal @ 700°F	.90	65 ft	66,535 ft	8.1897/ft	12,622	10.05	7.31
	Tube Brackets	.90	43 ea	52,776 ea	.04 ea	2,111	.42	.30
	Insulation	.95	23 ea	34,476 ea	.06 ea	2,164	1.74	1.25
	Miscellaneous Hardware	-	-	-	-	1,255	1.00	.72
	TOTAL ABSORBER					14,948	\$12.21	\$ 8.86

	NET	ANNUAL	\$/PANEL	\$/ft ²
SHELL	109,039	25,519	6 7.00	\$ 5.03
REFLECTION	82,680	28,082	27.36	16.05
SUPPORT	81,324	11,011	9.48	6.82
DRIVES	-	-	-	-
ABSORBER	6,611	16,580	22.21	8.40
TOTAL	278,743	471,192	61.05	44.30

A-13 CPC DOUBLE SIZE

4' x 8' = 32' sq = 2.97 m² projected area

MATERIALS ONLY

Component	\$/Panel	\$/m ²	Notes
<u>Shell</u>			
Frame	\$ 11.38	\$ 3.83	Varies with actual circumference
Bottom Pan	8.18	2.74	Varies as total area
Inter Pan	6.70	2.26	Varies as area
Foam	13.95	4.70	Varies as wall and bottom area
Discount 15%	(6.03)	(2.03)	-
Misc. Materials	1.75	.59	-
TOTAL	\$ 35.93	\$12.09	
<u>Reflector</u>			
Aluminum Coil	\$ 37.32	\$12.56	Varies as area
Brackets	1.96	.66	Varies as area
Cover Glass	9.14	3.08	Varies as area
Cover Gasket	3.64	1.22	Varies with circumference
Aluminum Selvedge	(.28)	(.09)	-
Discount 15%	(7.81)	(2.63)	-
TOTAL	\$ 43.97	\$14.80	
<u>Supports</u>			
Structural Steel Channels	\$ 10.91	\$ 3.67	Varies Discontinuously with area
Pads	.45	.13	Varies Discontinuously with area
Misc. Hardware	1.00	.34	-
Steel Selvedge	(.19)	(.06)	-
TOTAL	\$ 12.17	\$ 4.08	
<u>Drives</u>			
None			
<u>Absorber</u>			
Carbon Steel Tube	\$ 20.10	\$ 6.77	Varies with area
Tube Bracket	.84	.28	Varies with area
Insulator	3.48	1.17	Varies with area
Misc. Hardware	2.00	.67	-
TOTAL	\$ 26.42	\$ 8.89	
<u>GRAND TOTAL</u>			
Shell	\$ 35.93	\$12.09	
Reflector	43.97	14.80	
Supports	12.17	4.08	
Absorber	26.42	8.89	
TOTAL	\$118.49	\$39.86	

Materials Only

First Approximation - no change in labor since total projected areas are the same for the year's production.

A-14 CPC SUMMARY
CAPITAL INVESTMENT
(\$000)

Machinery and Equipment	\$11,860
Engineering @ 20%	2,372
TOTAL MACHINERY AND EQUIPMENT	\$14,232
Building 150,000 Sq. Ft. @ \$25/sq ²	3,750
Facilities @ 20% of Equipment	2,846
Landscaping, Roads, Docks, and Grounds @ 10% of Building	375
Architect and Planning @ 10% of Building	375
TOTAL BUILDINGS AND IMPROVEMENTS	\$7,346
LAND, 100 ACRES @ \$10,000/Ac	1,000
TOTAL CAPITAL INVESTMENT	\$22,578

CPC #6 STATEMENT FORMAT
(\$000)

46 Line No.		
10	<u>Direct Cost, Labor</u> (519 Operators)	\$ 3,324
11	<u>Direct Cost, Materials</u> (1,253,380 Panels Mer)	71,190
	Supervision (5 Supervisors @ \$20,000/Yr)	100
	Tooling	950
	Power (10,000 KWH @ \$.037/KWH)	370
	Freight (\$.01/lb on Raw Materials)	2,790
	Pension and Insurance @ 15% of Direct Labor	499
	Maintenance (10% of Capital Investment/Yr) in Machinery or Equipment	2,136
12	<u>Direct Cost, Factory Expense</u>	5,903
17	<u>Direct Product Cost</u> (Sum Lines 10, 11 and 12)	80,417
19	Manufacturing (5 Engineers @ \$15,000/Yr + 15%)	86
20	Engineering (2 Engineers @ \$20,000/Yr + 15%)	46
21	Marketing (2 Salesmen @ \$15,000/Yr + 15%)	34
22	Administration and General (Administration, Clerical and Security + 15%)	483
24	TOTAL OPERATING MANAGED COSTS	649
26	Depreciation @ 10%/Yr on Equipment	\$ 1,423
	5%/Yr on Building	367
		1,790
	Insurance and Taxes @ 2% of Capital Investment (Total)	452
	Holidays and Vacations (20 Days Average/Person)	512
27	TOTAL	964
28	Division Committed Costs (Sum Lines 26 and 27)	2,754
30	TOTAL OPERATING COSTS (Sum Lines 17, 24 and 28)	\$83,820
		(\$66.77/Panel)
		(\$47.90/m ²)

APPENDIX B
THERMODYNAMIC PROPERTIES OF METHANOL

THERMODYNAMIC PROPERTIES OF METHANOL

The thermodynamic properties used in the methanol cycle analysis follow:

- Liquid specific heat

$$C_l = 2.51 \text{ joules/gm-}^\circ\text{C at } 20^\circ\text{C}$$

$$C_l = 2.58 \text{ joules/gm-}^\circ\text{C at } 40^\circ\text{C}$$

These are taken from the International Critical Tables (20) and extrapolated linearly within the temperature range of interest.

- Liquid density

$$\rho_l = .8099 - .0009253T - .41 T^2 \times 10^{-6}$$

where ρ_l is in gm/ml and T in $^\circ\text{C}$. This is taken also from International Critical Tables (20).

- Vapor specific heat

$$C_p = 1.917 \text{ joules/gm-}^\circ\text{C between } 100^\circ\text{C and } 223^\circ\text{C, from}$$

International Critical Tables (20).

- Specific heat ratio

$$\gamma = 1.203, \text{ from } \underline{\text{International Critical Tables}} \text{ (20).}$$

- Critical constants

$$T_c = 240.0^\circ\text{C}$$

$$P_c = 59660.0 \text{ mm-Hg}$$

These were taken from Physical Properties of Chemical Substances, (21) Dow Chemical Company.

- Pressure-volume-temperature relation for vapor

The Berthelot equation was used:

$$PV = RT \left[1 + \frac{9}{128} \frac{P}{P_c} \frac{T_c}{T} \left(1 - 6 \frac{T_c^2}{T^2} \right) \right]$$

• Vapor pressure

The tabular data below for vapor pressure versus temperature were taken from Perry (22).

<u>Temperature(°C)</u>	<u>Vapor Pressure (atm)</u>
80	1.764
100	3.452
120	6.255
140	10.63
160	17.11

The vapor pressure at any given temperature is interpolated (or extrapolated) from the two nearest absolute temperature points (T_1, T_2) by the formula:

$$P = P_1 \left(\frac{P_2}{P_1} \right)^X$$

where $X = \frac{T_2}{T} \left(\frac{T - T_1}{T_2 - T_1} \right)$.

This is equivalent to assuming the exponential relation

$$P = AC - \frac{B}{T}$$

where A and B are chosen to fit the two reference temperature points.

• Latent heat

Straight line interpolation was used with the following tabular data from Perry (22).

<u>T(°C)</u>	<u>L($\frac{\text{cal}}{\text{gm}}$)</u>
64.7	262.79
100	241.29
160	193.51
200	148.12
220	109.89
240	0.0