(10) 2/3/84

SERI/TR-253-1662
DE84000024

November 1983

Low-Temperature IPH Parabolic Troughs: Design Variations and Cost-Reduction Potential

Randy C. Gee

SERI

Solar Energy Research Institute
A Division of Midwest Research Institute

1617 Cole Boulevard
Golden, Colorado 80401

Operated for the
U.S. Department of Energy
under Contract No. DE-AC02-83CH10093

Printed in the United States of America
Available from:
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161
Price:
Microfiche A01
Printed Copy A03

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Department of Energy, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

SERI/TR-253-1662
UC Categories: 59b, 59c, 62a
DE84000024

Low-Temperature IPH Parabolic Troughs: Design Variations and Cost-Reduction Potential

Randy C. Gee

November 1983

Prepared under Task Nos. 1270.40 and 1387.30
WPA No. 281-82

Solar Energy Research Institute
A Division of Midwest Research Institute

1617 Cole Boulevard
Golden, Colorado 80401

Prepared for the
U.S. Department of Energy
Contract No. DE-AC02-83CH10093

PREFACE

This work was supported by the U.S. Department of Energy as part of SERI's effort to identify overall solar thermal system cost-reduction opportunities.

The author would like to thank Barry Butler of the Solar Thermal and Materials Research Division at SERI for encouraging the investigation of low-cost concentrating collectors and L. M. Murphy for supporting this work and providing valuable comments and suggestions. The report was reviewed by collector and solar system designers at Sandia National Laboratories, Albuquerque, who are actively involved with the development of parabolic trough technology. Thanks are expressed to these reviewers and to SERI reviewers Bim Gupta, Charles Kutscher, and E. Kenneth May.

Approved for

SOLAR ENERGY RESEARCH INSTITUTE

Charles F. Kutscher for JPT

John Thornton, Chief
Thermal Systems and Engineering
Branch

John P. Thornton for B. Butler

Barry Butler, Manager
Solar Thermal and Materials
Research Division

SUMMARY

Parabolic trough concentrating collectors have improved greatly in performance and reliability since the U.S. Department of Energy began its solar thermal program. The development of components has generally been tailored to meeting instantaneous collector performance goals at an operating temperature of 315°C (600°F). Although this strategy has resulted in advanced component designs and excellent collector performance at 315°C, it has also necessitated rather stringent and expensive component design requirements. Near-term parabolic trough collector systems are too expensive to compete widely with conventional fossil-fuel systems.

This report describes how collector costs can be reduced by means of lower-temperature designs. A low-temperature design strategy is supported by the fact that industrial process heating energy use at below 150°C (300°F) is over five times greater than energy use at from 200° to 315°C (400°F to 600°F). Component designs for lower temperatures are considered, including less rigid, lighter-weight concentrators; alternative concentrator constructions; larger receivers; and multiple-row drive systems.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

SECTION 1.0

INTRODUCTION

Parabolic trough concentrating collectors have been actively developed since 1973. Until recently, this development work has focused on improving the collectors' performance and on demonstrating the technical feasibility of the technology. Development efforts by collector manufacturers and DOE-funded national laboratories (principally Sandia National Laboratories in Albuquerque) have resulted in considerable improvements in collectors' performance. Parabolic troughs are now being manufactured that have optical efficiencies up to 80% (Harrison 1981). Several field experiments, funded by the U.S. Government (first by ERDA and then DOE), have also provided valuable field experience and identified areas where improvements were needed (Kutscher 1981). The final cycle of industrial process heat (IPH) field experiments has just been completed, and these systems are now beginning operation.

As this initial technology development stage nears completion, the cost-effectiveness of parabolic trough concentrating collectors has emerged as the central issue. The final two DOE-sponsored IPH field experiments that have just begun operation each cost over $50 per unit of collector aperture area. Based on the analysis of Edelstein et al. (1982), it is clear that parabolic trough systems that cost this much are generally not competitive with conventional fossil-fuel systems. Major cost reductions are necessary before parabolic trough solar energy systems can be widely competitive with conventional fuels. The current costs of parabolic trough systems generally limit their cost-effective use to very special applications in areas with high direct-normal insolation. These special applications [e.g., competing with an unusually expensive alternative fuel or inefficient heating equipment (Hooker 1980)] are not numerous and represent only a small fraction of total U.S. energy use.

Some cost reductions in parabolic trough systems are expected to occur as system installation costs are reduced, larger quantities of collectors are produced, and more efficient manufacturing techniques are implemented. Recently, two DOE-funded programs were undertaken to support such cost reductions. Contracts for the development of mass-producible line-focus tracking concentrating solar collectors and their components and subsystems were recently completed by several parabolic trough manufacturers. Also, the Modular Industrial Solar Retrofit (MISR) program was initiated in 1981. The MISR program is principally aimed at reducing system installation costs through modular system design. These programs have resulted in improved designs but few (if any) overall system cost reductions. Cost reductions based on learning-curve approaches are projected, but the magnitude of the cost reductions needed suggests that a departure from currently used parabolic trough designs is now called for.

Most commercially available parabolic troughs suitable for solar IPH systems have been designed to meet instantaneous performance goals for an operating temperature of 315°C (600°F). This strategy has led to rather stringent (and expensive) component design requirements. However, most IPH energy use below about 540°C (1000°F) occurs at very low temperatures. In fact, the greatest use occurs at temperatures of 100° to 150°C. This report shows that component design requirements can be relaxed for lower operating temperatures. Hence, for this lower temperature range, several cost-saving design changes for parabolic trough collectors are possible. These design changes, with an appropriate research and development effort, may provide significant cost reductions and greatly enhance parabolic trough system economics.

In the following section, 2.0, we describe two fundamental characteristics of parabolic troughs designed for IPH applications. First, IPH temperature requirements are briefly described and data are presented that indicate that a large share of IPH thermal energy is required at temperatures below 150°C. Next, we describe cost goals for IPH collector systems set by DOE. These goals indicate the extent to which collector system cost reductions are necessary before solar systems can be cost-effective in IPH applications. Section 3.0 suggests a strategy for designing parabolic troughs specifically for low-temperature IPH applications. A graphical analysis technique illustrates the sensitivity of parabolic trough collector designs to operating temperatures, and opportunities for cost-savings through lower temperature designs are presented. The conclusions of our study are contained in Sec. 4.0.

SECTION 2.0

APPLICATION-GOVERNED CHARACTERISTICS OF AN IPH PARABOLIC TROUGH

Two important technical and economic characteristics of a solar collector used to supply thermal energy to industry are governed by characteristics of the industry itself. First, the operating temperature of the industrial process largely determines the operating temperature of the solar collector, which in turn has an impact on the design of the collector. Second, the cost of the fossil fuels used for industrial process heat defines the value or allowable cost of a solar collector in the marketplace.

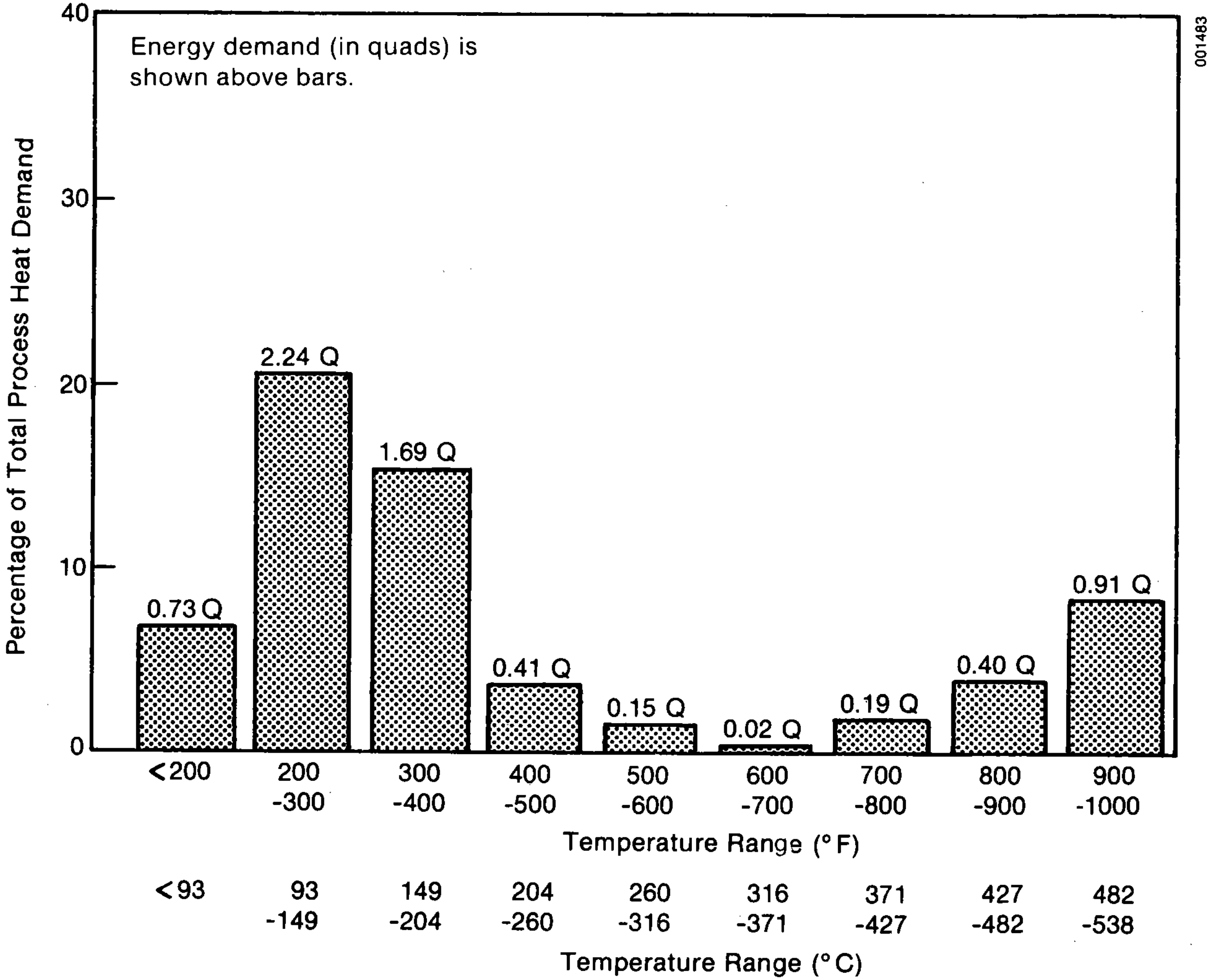
2.1 IPH END-USE TEMPERATURE REQUIREMENTS

Most parabolic trough collectors and their components have been developed to meet instantaneous performance goals for an operating temperature of 315°C (600°F). This operating temperature was designated a suitable design point because parabolic trough systems could then be used in a number of applications, including electric power generation, industrial process heating, and solar heating and cooling. However, industrial process heating has emerged as the largest potential application of parabolic trough technology. Therefore, it is now more appropriate to examine industrial energy use alone to establish a suitable parabolic trough design-point temperature.

Of the percentage of total U.S. energy demand devoted to industry, about 70% is direct thermal energy. Direct thermal energy is used for a variety of industrial processes over a wide range of temperatures. A recent, detailed analysis of the IPH market (Krawiec 1981) provided, for the first time, reliable data on industrial energy demand, end uses, and end-use temperature levels. An end-use temperature distribution was projected for the year 1990 and is shown in Fig. 2-1. Note that a great amount of energy is needed at the lower end of the range. In fact, about 44% of all process heat will be used at temperatures below 200°C (400°F). IPH energy demand at below 200°C is over eight times greater than at the higher temperature range, 200°C to 315°C (400°F to 600°F). The temperature range 100°C to 150°C is particularly energy-intensive, accounting for over 2.2 quads* of IPH energy use.

Examining lower-temperature IPH applications in more detail, we find that the large quantities of energy needed for process heating below 150°C are utilized principally to heat water or air or to produce low-pressure saturated steam. Hot water is needed for processes such as washing, bleaching, cooking, and anodizing. Hot air is used widely for drying processes; some of the large number of materials that require drying include lumber, food, paper products, paint, and printed matter. A breakdown of IPH energy use below 150°C is given in Fig. 2-2. The greatest use of low-temperature process heat is for low-pressure saturated steam. Steam is used for such applications as pasteurization, cleaning, reactor vessel heating, and food processing. Figure 2-3 shows IPH energy use below 150°C by industrial sector. Three industries utilize almost 75% of the total process energy below 150°C: food, paper, and chemical products.

*1 quad = 10^{15} Btu.



SOURCE: Krawiec 1981.

Figure 2-1. Projected Distribution of IPH Demand at Below 1000°F, by Temperature Range (Year 1990)

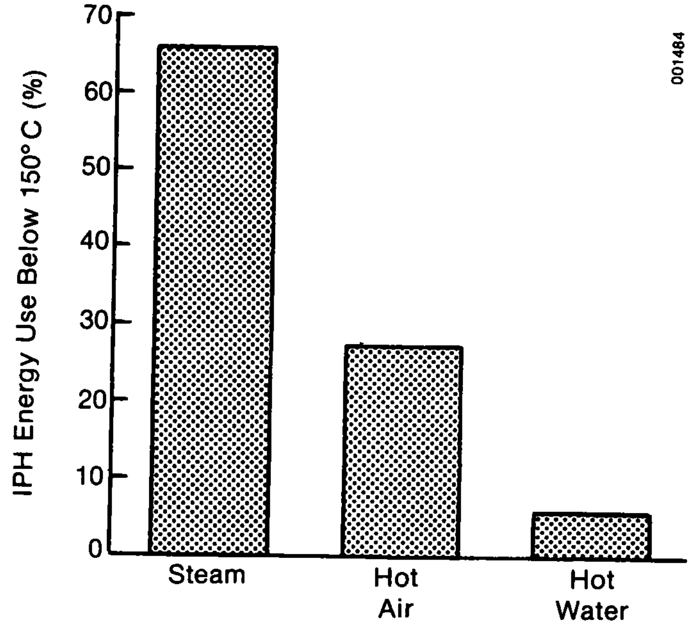


Figure 2-2. IPH Energy Use at Below 150°C, by End-Use Fluid

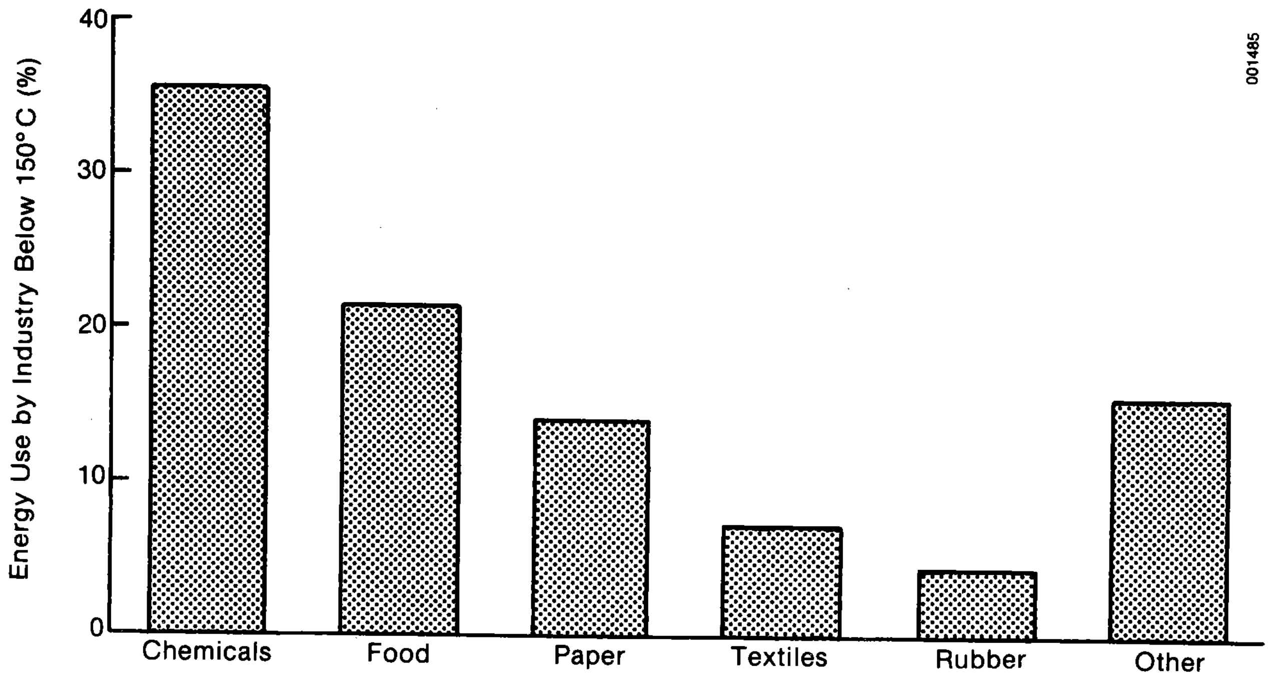


Figure 2-3. IPH Energy Use at Below 150°C, by Industry

For the operating temperature range that is usually considered appropriate for parabolic trough applications (up to 315°C), almost 90% is below 200°C and about 60% is below 150°C. Hence, the historical design-point temperature of 315°C is not well matched to industrial process heat applications, and a lower one is more appropriate.

The importance of using a much lower design-point operating temperature is twofold. First, a lower design-point temperature permits a significant relaxation of the design requirements of a parabolic trough. Relaxed collector design requirements allow us to design and use collector components much different from, and potentially lower in cost than, those in current use. Second, a lower collector operating temperature reduces thermal losses from the collector to the environment, increases collector efficiency, and results in greater annual energy collection. This effect, by itself, improves the cost-effectiveness of the solar system. In Sec. 3.0, we describe in more detail how the low-temperature characteristic of the IPH market can lead to significant cost reductions.

2.2 COST GOALS FOR IPH PARABOLIC TROUGHS

An interlaboratory Solar Thermal Cost Goals Committee (STCGC) was funded by the Department of Energy's Solar Thermal Energy Systems Division to recommend value-based cost and performance goals for solar thermal systems (Edelstein 1980). The objective of the STCGC was to establish cost goals to aid DOE's solar research and development programs. Value-based goals are set according to the value of the energy produced by the technology. Hence, if these cost goals are met, the technology stands a very good chance of being applied to the IPH sector. The method chosen for setting these value-based goals was to set the life-cycle cost of the solar thermal system equal to the levelized cost of energy (from the competing fossil-fuel-burning system) that is displaced by the solar system. Because both solar system performance and the costs of fossil fuels vary according to geographic location, value-based cost goals were defined as a range rather than as one unique value.

A computerized analysis was employed to evaluate the impact of geographic location on solar thermal system value (Flowers 1982). This analysis accounted for the geographic variations of both fuel price and solar system performance. Estimated 1990 fuel prices for coal, natural gas, distillate oil, and residual oil for the United States as a whole are given in Table 2-1, as well as the assumed fuel escalation rates beyond 1990 for the four fuels. The fossil-fuel-burning system is assumed to have an efficiency of 70%.

Solar system performance depends on geographic location, principally because of geographic variations in average solar irradiation. This variation with location was accounted for by the distribution of direct-normal irradiance throughout the nation. Solar system life-cycle cost methods require estimates of a number of economic parameters, many of which are uncertain. The economic parameters used by the STCGC are listed in Table 2-2. These economic assumptions are consistent with the scheduled cessation of federal solar system tax credits in December 1985.

Table 2-1. Fuel Costs and Escalation Rates

Fuel	1990 Price (1981$)	Real Fuel Escalation Rate (1990 to 2000)
Natural gas	$6.65/MBtu	2.8%
Distillate oil	$9.40/MBtu	3.9%
Residual oil	$7.90/MBtu	2.7%
Coal	$2.40/MBtu	0.8%

Table 2-2. Baseline Economic Parameters Used To Set Cost Goals

Investment tax credit	10%
Depreciable life	5 yrs
Operation and maintenance	2% of capital cost
Property tax and insurance	2% of capital cost
General inflation rate	6%
System lifetime	20 yrs
Fixed charge rate	0.277
Loan fraction	0

The relationship between national solar IPH potential and initial solar system cost is shown in Fig. 2-4. IPH potential (quads per year) is shown as a function of total installed solar thermal system cost, in 1980 dollars, per unit of collector system aperture area. A variation in annual system efficiency of 50% to 65% is shown. This system efficiency is defined with respect to annual direct-normal irradiation. Defined in these terms, a parabolic trough system efficiency of 50% is quite high (even at lower operating temperatures) and represents an ambitious 1990 performance goal.* Note that, for an assumed system efficiency of 50%, the total solar thermal system installed cost must be below $215/m^2 ($20/ft^2) before solar IPH systems can begin to compete with

*The higher efficiency curves shown in Fig. 2-4 (55%-65%) may be more applicable to other solar thermal technologies, such as parabolic dishes or central receivers.

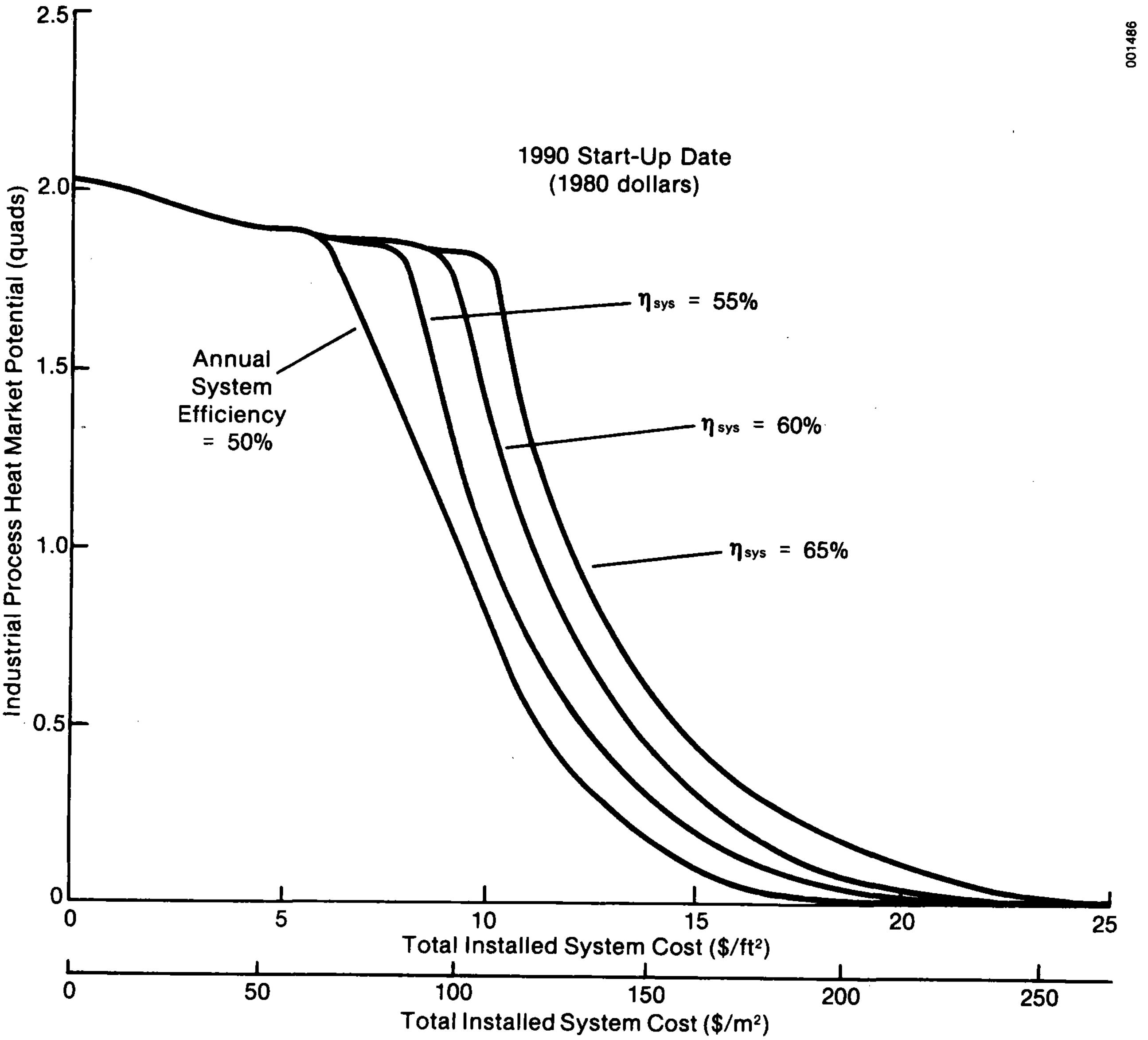


Figure 2-4. IPH Potential vs. Total System Cost

Source: Edelstein 1982.

conventional fossil fuel systems.* To achieve a significant penetration into the industrial sector, (>1 quad),** system costs must be below about $\$100/m^2$ ($\$10/ft^2$).

Because pumps, piping, heat exchangers, and other equipment contribute a sizable amount to total solar system costs, installed collector costs represent only a part of the total solar system cost. The cost goals formulated by the STCGC assumed that 50% to 60% of system costs are associated with the collector. With this assumption, installed parabolic trough costs must be below about $\$60/m^2$ ($\$6/ft^2$) to achieve a significant penetration (>1 quad) into the industrial sector.

*In the near term, parabolic trough systems may be competitive with some conventional fossil-fuel systems because of innovative financing arrangements and tax credits (Dickinson 1980).

**A IPH penetration of 1 quad would require approximately 300×10^6 m^2 of collector area.

SERI

SECTION 3.0

THE IMPACT OF OPERATING TEMPERATURE ON PARABOLIC TROUGH DESIGN

In Sec. 2.0, we showed that IPH applications require an enormous amount of energy at temperatures below 150°C. To be competitive with conventional fossil-fuel boilers, solar thermal systems must cost significantly less than they do now. In this section, we describe how the low-temperature characteristic of the IPH sector can make significant cost reductions possible for parabolic troughs, to help meet the solar thermal system cost goal.

In Sec. 3.1, we describe how and why parabolic trough design and performance are affected by design-point temperature and how annual performance is impacted by the design characteristics of the trough. Section 3.2 presents three cost-saving component design changes that are especially suited to low-temperature designs.

3.1 DESIGN SENSITIVITY TO OPERATING TEMPERATURE

The optimum design of a parabolic trough concentrating collector depends on the intended end-use temperature of the collector. Design aspects of all collector components are, to some degree, affected by the collector's operating temperature. This section shows, in general terms for the component designer, how the design of individual components affects overall collector performance as a function of operating temperature.

To assess design alternatives for individual collector components properly, an accurate measure is needed of how the alternatives impact the performance of the entire collector. Often, instantaneous collector efficiency is this measure of performance. However, instantaneous efficiency can be misleading because it underestimates the impact of thermal losses and off-design-point operation (Gee and Murphy 1981). Instead, annual collection efficiency should be used.

To illustrate some of the important temperature-dependent design characteristics of a parabolic trough, annual collector efficiency is shown as a function of heat loss in Fig. 3-1. Annual collection efficiency is defined here as the ratio of the annual collected energy to the annual beam insolation falling on the collector aperture from sunrise to sunset. (Efficiency defined in these terms is about a factor of 1.3 higher than efficiency expressed in terms of direct-normal irradiation.) This graph shows how annual efficiency varies with receiver heat-loss rate for three parabolic troughs that are identical except for receiver size. Indicated along the x-axis are the approximate absorber-tube temperatures that correspond to the heat-loss rates for a typical state-of-the-art trough receiver. Three receiver sizes are shown, corresponding to geometric concentration ratios* of 15, 20, and 25. For each concentration ratio, a range in collector optical precision is shown. This range is the same for all concentration ratios. Qualitatively, optical

*The geometric concentration ratio is defined as $C = W/(\pi d_{abs})$.

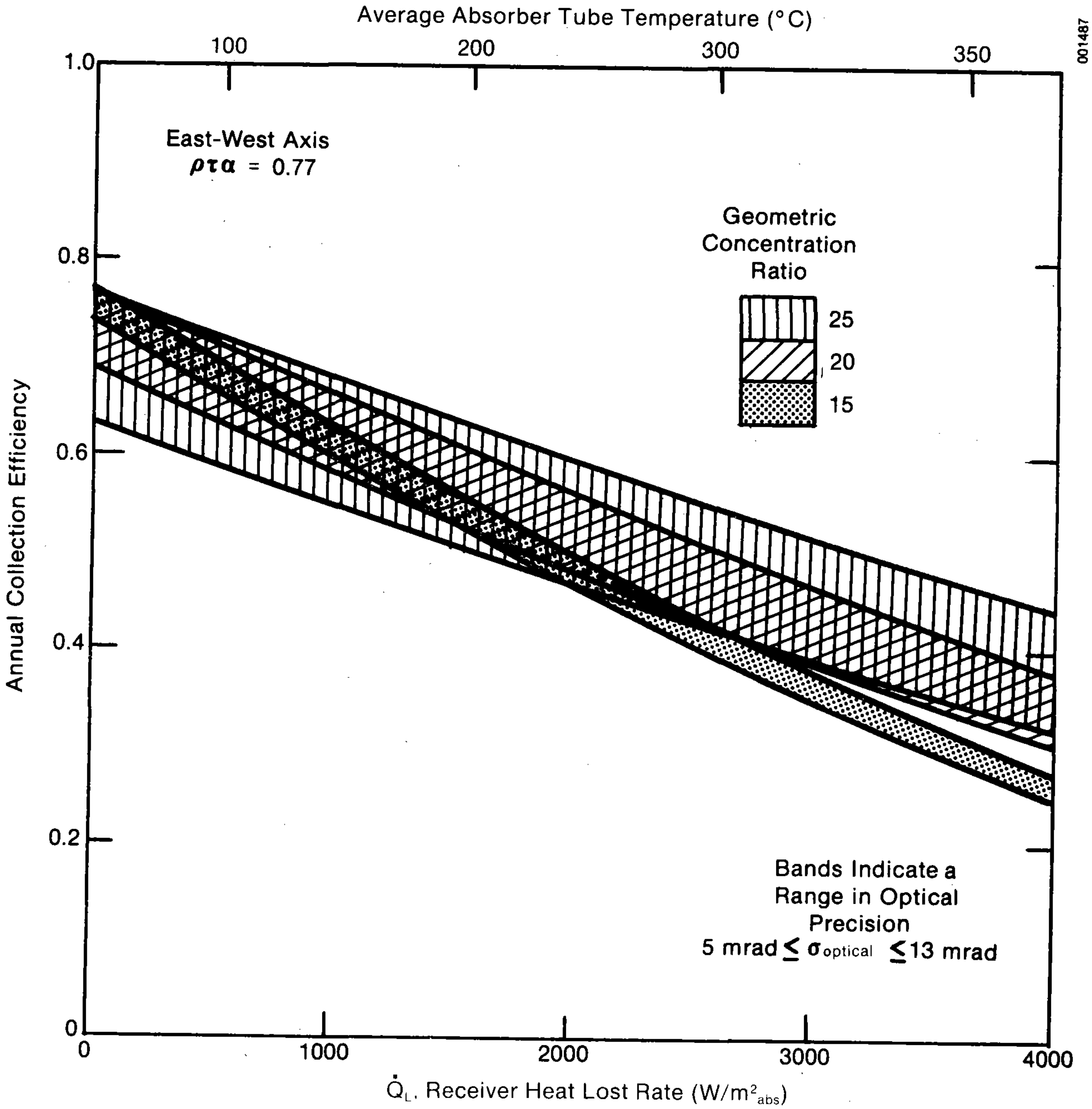


Figure 3-1. Sensitivity of Trough Performance to Heat Loss and Optical Accuracy

precision ranges here from excellent to mediocre. Quantitatively, this optical error ranges from 5 to 13 mrad. The optical error total is the root-mean-square (rms) angular spread of reflected irradiation caused by all optical imprecision (as discussed later in this section).

Note that the importance of optical precision in annual efficiency depends strongly on the concentration ratio of the collector. Collectors with high concentration ratios are very sensitive to optical precision, whereas those with low concentration ratios are much less sensitive. This is so because of the interrelationship of concentration ratio and optical precision in defining the optical intercept factor of the collector. For high concentration ratios, the target that the receiver presents to the concentrator is relatively small. If optical precision is high, most of the reflected sunlight will hit the receiver. If the optical precision is significantly lowered (i.e., the rms contour error is increased), much of the reflected sunlight will miss the receiver and collector performance will degrade significantly. At low concentration ratios, the target the receiver presents to the concentrator is relatively large, and most of the reflected sunlight will be intercepted by the receiver even if optical precision is not great.

While the selection of the concentration ratio of a parabolic trough is related to the optical precision of the concentrator, it also depends on the heat-loss rate of the receiver. Figure 3-1 shows, as one would expect, that at high receiver heat- loss rates, higher concentration-ratio collectors perform best. At lower receiver heat-loss rates, lower concentration-ratio collectors, even with poorer optical precision, can perform as well or better than collectors with high concentration ratios.

The annual efficiency of a parabolic trough is further quantified as a function of collector component characteristics in Figs. 3-2 and 3-3. These figures show collector annual efficiency in a more general way than Fig. 3-1 does. First, concentration ratio is no longer fixed. Instead, the concentration ratio is varied such that it is at its optimum value for each location on the figure. Hence, for each mix of collector design characteristics, the designated concentration ratio of the trough maximizes annual efficiency. In essence, Figs. 3-2 and 3-3 automatically incorporate the performance impact of concentration ratio. Second, annual efficiency is shown as a function of three independent variable groupings that can be computed easily from the properties and characteristics of the individual components of a given parabolic trough. The figures yield quite accurate results over a wide range of climates (Gee et al. 1980). The annual collector calculation procedure on which the figures are based is summarized in the Appendix.

The first variable grouping, $\rho\tau\alpha$, is simply the product of concentrator hemispherical reflectance, receiver glazing transmittance, and receiver-tube selective surface absorptance, all at normal incidence. The next variable is the receiver heat-loss rate, Q_L, at the operating temperature of the trough. The final variable is the effective total optical error at normal incidence that results from imperfect tracking and concentrator optics.

$\rho\tau\alpha$ = (hemispherical reflectance) x (receiver glazing transmittance at normal incidence) x (receiver-tube selective surface absorptance at normal incidence)

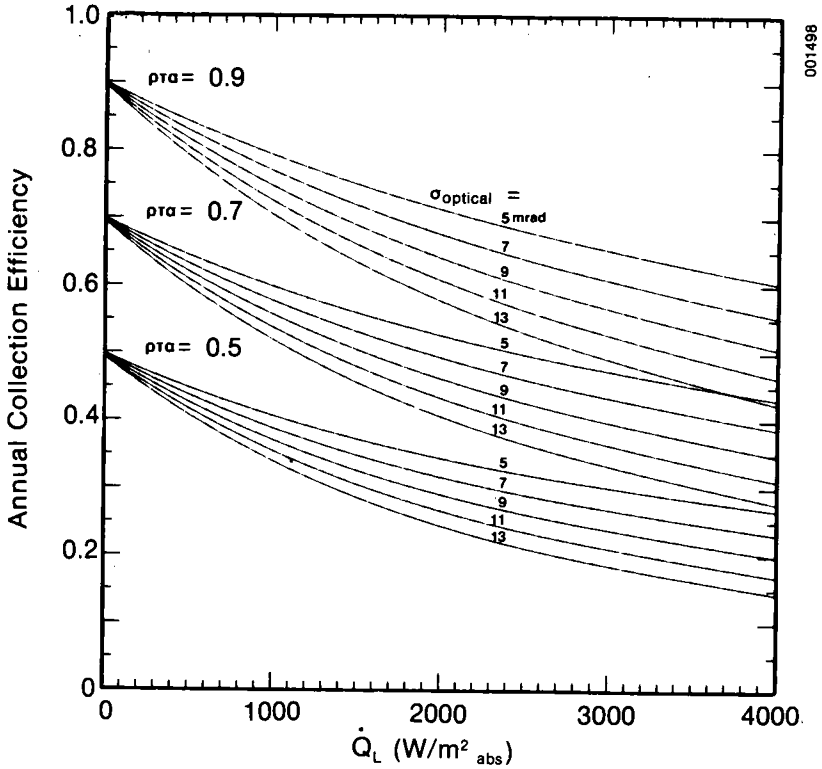


Figure 3-2. Annual Collection Efficiency of East-West Rotational Axis Parabolic Trough vs. Receiver Heat-Loss Rate

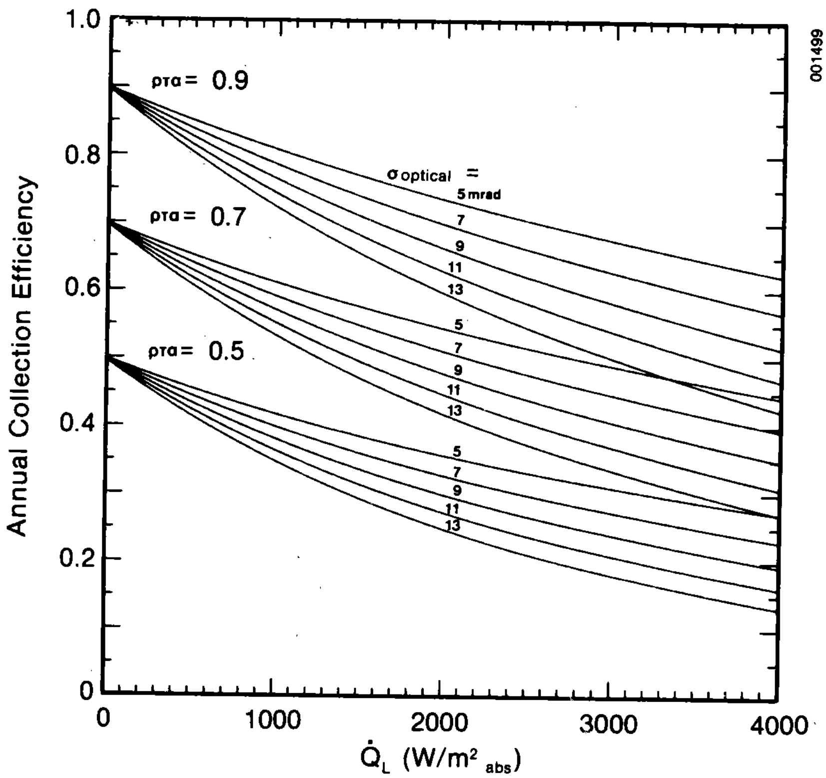


Figure 3-3. Annual Collection Efficiency for North-South Rotational Axis Parabolic Trough vs. Receiver Heat-Loss Rate

$\dot{Q}_L = U_L (T_{abs} - T_{amb})$ = heat-loss rate from receiver in watts per square meter of absorber tube area

$\sigma_{optical} = (4\ \sigma^2_{con} + \sigma^2_{track} + \sigma^2_{disp} + \sigma^2_{spec})^{1/2}$, expressed in milliradians (note that sun size is not included in this term) ,

where

σ_{con} = rms angular deviation of concentration from perfect parabola (slope error)

σ_{track} = rms angular spread due to sun tracking error (effective annual value)

σ_{disp} = equivalent rms angular spread, which accounts for the imperfect placement of the receiver

σ_{spec} = rms angular spread of reflected beam due to imperfect specularity of reflector.

Note that receiver thermal loss is expressed in terms of heat loss per unit of absorber tube surface area. Estimates of U_L for various parabolic trough receiver designs are given in Fig. 3-4. Alternatively, a value of $\dot{Q}_L$ can be easily calculated for any parabolic trough, given its instantaneous efficiency equation. The receiver thermal loss rate (in W/m^2_{abs}) is simply the product of three quantities: the overall U-value of the collector, the average ΔT of the collector, and the trough's geometric concentration ratio.

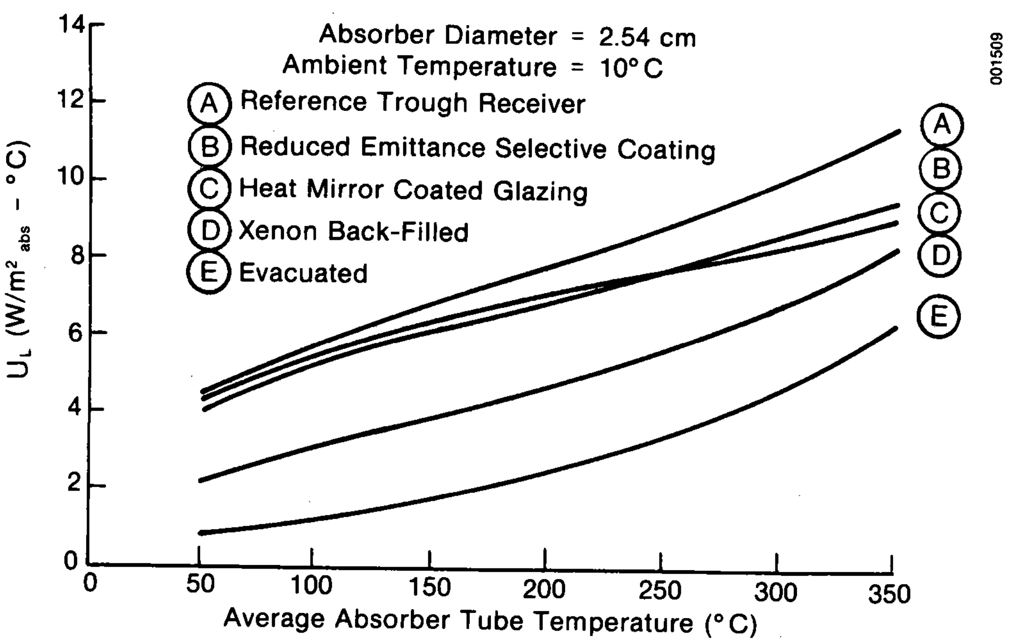


Figure 3-4. Heat-Loss Coefficients vs. Temperature

With these graphs, the performance impact of various design alternatives for a trough component can be quickly evaluated. Also, the sensitivities of parabolic trough performance to the three variable groupings (the optical efficiency term $\rho\tau\alpha$, the optical surface accuracy $\sigma_{optical}$, and the receiver heat-loss rate $\dot{Q}_L$) can be readily seen and are described briefly below.

Note that annual collection efficiency decreases significantly as receiver heat-loss rate (or operating temperature) increases. Typical parabolic trough peak instantaneous efficiencies suggest that thermal loss has much less significance. The significance of heat loss to peak instantaneous efficiency is quite small because instantaneous efficiencies are determined when insolation levels are high and incidence angles, low. Over a year, the influences of lower average insolation and off-peak performance result in greater sensitivity to thermal loss.

The impact of the $\rho\tau\alpha$ term is relatively constant for all heat-loss levels. Increasing concentrator reflectance, receiver transmittance, or absorptance by 10% is shown to result in about a 10% increase in annual efficiency for all values of $\dot{Q}_L$.

As we discussed earlier, the impact of optical precision on trough performance is highly dependent on the concentration ratio of the collector. Figures 3-2 and 3-3 show this dependence because the concentration ratio optimization is built into the graphs. The sensitivity of parabolic trough annual efficiency to $\sigma_{optical}$ is high for collectors designed and operated at high temperatures (and, therefore, high $\dot{Q}_L$ values). Hence, a normal incidence optical error budget ($\sigma_{optical}$) of 7 mrad has been established by Sandia National Laboratories for efficient high-temperature operation (Bergeron et al. 1980). However, for low-temperature operation (low $\dot{Q}_L$ values), the significance of $\sigma_{optical}$ is minimal. For example, a $\sigma_{optical}$ increase of 4 mrad decreases annual performance by 8% at 315°C but only 4% at 150°C*. Thus, at low temperatures, concentrators with less than premium optical characteristics could still exhibit good collector performance. If less optically precise parabolic trough collectors can be built with significant cost savings, the lower cost will greatly offset the small performance penalty.

3.2 LOW-TEMPERATURE DESIGNS

In Sec. 2.2, we pointed out that large cost reductions in parabolic troughs will be necessary before a large solar IPH market potential can exist. Achieving such significant cost reductions will probably require departures from conventional parabolic trough design. The previous section showed that at low operating temperatures, collector optical precision requirements may be relaxed, and the performance penalty associated with increased optical errors is much lower than at high operating temperatures. Some component design changes that may be appropriate for low-temperature IPH parabolic troughs are discussed briefly in this section. While some additional information about these parabolic trough component designs is available from the references

---

*This example assumes that the collector concentration ratio has been optimized for each operating temperature.

cited in the text, a great deal of further work is needed to investigate their performance and costs.

3.2.1 Lightweight Concentrators

The parabolic-shaped concentrator of a parabolic trough is the single most expensive component of the collector. Almost 50% of the total current cost of parabolic troughs is associated with the concentrator (Gee and Murphy 1981). Reducing the cost of the concentrator is, therefore, likely to have a significant impact on the total cost of the collector.

Current parabolic concentrators are designed to be optically precise. An rms concentrator contour error of 2.5 mrad is considered the optical accuracy goal for troughs (Bergeron et al. 1980). This design value was based on the collector's delivering thermal energy at a temperature of 315°C (600°F). At that temperature, as shown in Sec. 3.1, precise optics are very important to high collector efficiency. Parabolic concentrators have been designed and fabricated that have successfully met the 2.5 mrad optical accuracy requirement (Champion 1980). However, these concentrators generally weigh 20-30 kg/m^2 (4-6 lb/ft^2) and do not appear to be capable of meeting the installed cost goal (including the receiver, pylons, foundations, controls, and drive assemblies) of $\$60/m^2$ ($\$6/ft^2$).

Designing for lower temperatures allows us to use lighter, less rigid concentrators because concentrator optical accuracy is not as critical at below 150°C as it is at 315°C or above. This increased tolerance for optical error occurs chiefly because, as shown in Sec. 3.1, lower concentration-ratio collectors can be used at lower temperatures without greatly increasing thermal losses.

Many concentrator designs have been constrained further by the requirement that the concentrator's structure must be sufficiently stiff to protect a thermally sagged glass reflector from fracture caused by gravity and wind-induced deformation of the concentrator (Murphy 1982). As a result, a concentrator flexural rigidity of 5.64×10^4 N-m (0.5×10^6 lb-in.) is considered to be the necessary design value for stiffness (Reuter 1980). However, if a thin polymer reflective film (or strengthened glass) is used, the reflector fracture constraint is removed, because polymer films and strengthened glass can be flexed. Thus, concentrator flexural rigidity can be significantly reduced below 5.64×10^4 N-m. In fact, for a concentrator utilizing a polymer reflector, flexural rigidity can be reduced by over an order of magnitude (below 5.64×10^3 N-m) and the concentrator contour error will increase by less than 2 mrad in a 30-mph wind (Murphy 1981).

Large reductions in the design value of concentrator flexural rigidity and larger tolerances in concentrator optical accuracy allow us to use much lighter-weight concentrators; this can result in significant cost savings. These cost savings will be realized with lightweight concentrators first because materials and fabrication costs will decrease as concentrator structure size decreases. Second, installation costs will drop because lightweight concentrators are easier to handle and install. Third, a lower flexural ridigity requirement allows us a wider choice of concentrator materials and designs. While slab and laminate constructions were found to be unsatisfactory (Reuter 1981) using a flexural rigidity requirement of 5.64×10^4 N-m, such constructions are acceptable if we use a lower design value for flexural rigidity. Designing to a lower stiffness allows us to use lower-cost materials, construction, and fabrication techniques that would otherwise be inappropriate.

Ultimately, determining the optimal stiffness of the concentrator involves a cost/performance tradeoff. As the concentrator's stiffness is reduced, the resulting cost savings should be weighed against the decrease in performance that occurs as concentrator slope errors increase. A thorough analysis of this cost/performance trade-off is a large undertaking because it depends on operating temperature, requires cost analysis of a variety of concentrator materials and constructions, and requires an accurate prediction of concentrator deformation under wind and gravity loads. Further, the trade-off also depends on the geometry of the modules, the length of the drive strings, and the manner in which the collector rows are driven. Also, the concentrator must be sufficiently stiff to ensure survival under peak wind loads. Although a detailed cost/performance optimization has not yet been done, the performance side of this optimization can be greatly enhanced by using Figs. 3-2 and 3-3; they provide for the transformation of optical errors into annual performance penalties.

3.2.2 Multiple-Row Drive Systems

Parabolic trough collectors are usually arranged in a number of parallel rows to form a collector field. Conventionally, each row is formed from a number of drive strings, each controlled and driven independently of the others to maintain precise tracking. Each drive string requires its own drive system, sun tracker, and local collector. The expense of drive and control hardware is considerable: drive and control costs amount to about 23% of the total FOB collector cost of an average parabolic trough (Gee and Murphy 1981). Additionally, each local controller requires power and control wiring.

An opportunity exists to reduce these parabolic trough drive and control costs significantly by mechanically ganging and driving multiple rows together. If several rows can be driven together, the total number of drives, sun trackers, controllers, and wiring for all of them can be reduced. Initial costs will be lower, and installation and maintenance costs will also be reduced.

Multiple-row drive systems are particularly attractive for parabolic trough IPH systems, which usually require a large collector field that is typically configured as a number of parallel rows. Because the rows are parallel, they all track at essentially the same angle at the same time during the day. This arrangement readily lends itself to a ganged drive. Also, parabolic trough rows can be placed quite close to each other. Row-to-row spacings of just 4 m can be used for typical east-west-oriented troughs (i.e., 2-m aperture width) with a minimum of shading loss (Kutscher et al. 1982). Except at very low latitudes, north-south-oriented troughs should be spaced farther apart.

Although a number of multiple-row drive systems can be conceptualized, the rim-drive multiple-row drive system is the one that has been investigated by

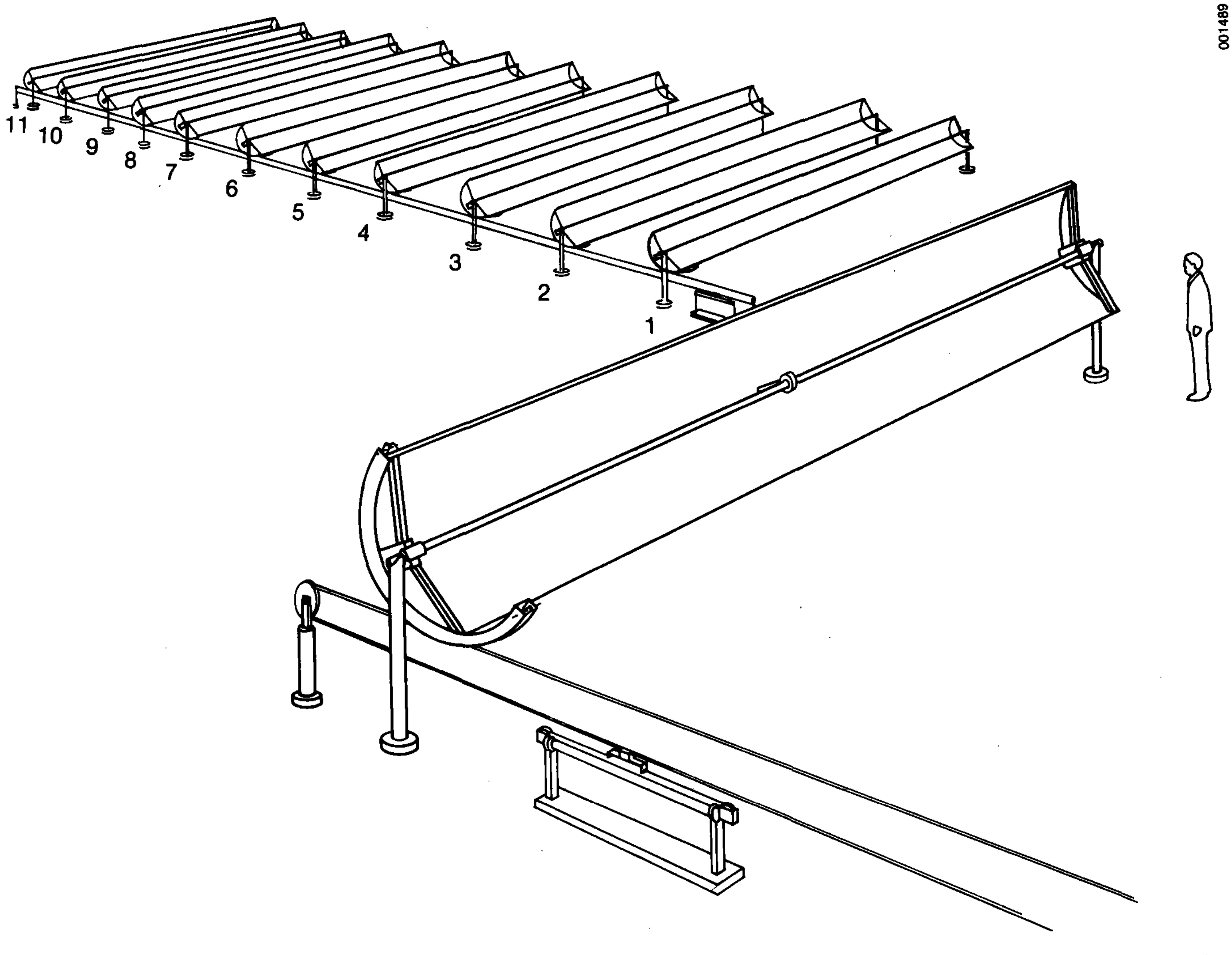


Figure 3-5. Rim-Driven Parabolic Trough System

SERI. With this system, multiple rows are coupled with wire-rope drive cables as shown in Fig. 3-5. As the master collector row tracks the sun, the cable along its rim wraps and unwraps. Adjacent "slave" rows are likewise forced to wrap and unwrap; that is, they follow the master row.

In this drive system, the collector's wind and gravity loads are transferred to the drive cable and tend to elongate that cable. These elongations result in row-to-row tracking errors that accumulate from row to row. Cable elongation from row 1 to row 2 (see Fig. 3-5) not only introduces tracking error to row 2 but also to subsequent rows (row 3 and so on). Further, if the cable connecting row 2 to row 3 elongates by the same amount as the row 1 to row 2 cable (as would be expected, since all collectors experience nearly the same loads), row 3 will exhibit twice the tracking error of row 2. Again, this cable elongation is passed on to all subsequent rows. Hence, this accumulation effect causes increasingly larger tracking errors for each row away from the master row. For example, a tenth row would have 47 times the tracking error of the second row.

Drive-cable elongations must be minimized in order to keep tracking errors to a level where performance is not greatly diminished. A tracking error of 6 mrad (0.4^{o}) results in a performance loss of only 2.5% for a collector with a geometric concentration ratio of 20 (Treadwell 1981). Collectors having lower concentration ratios will yield an even smaller performance penalty and those at higher concentration ratios will yield larger penalties. Treadwell does not report results for a lower concentration ratio, but does for a higher one. At a concentration ratio of 25, the performance loss associated with a 6-mrad tracking error is 4.5%. While further development of the concept is needed, recent data obtained from a prototype rim-drive system indicate that up to 15 rows can be driven within a 6-mrad tracking error budget (Murphy and Gee 1982).

Initial cost comparisons of this multiple-row drive system to a conventional drive system are encouraging; up to 80% of conventional drive system costs may be saved. A multiple-row rim-drive system has an additional attribute: the torsional stiffness requirement of the concentrator module is greatly reduced compared with conventional concentrator modules. The torsional stiffness of the concentrator can be reduced because the concentrator's support arms and end frames (along which the cable is wrapped) provide the necessary torsional rigidity. A structural analysis of this concentrator/drive arrangement, which can eliminate the expense and weight of the torque tube behind a conventional concentrator, is described in a recent report (Murphy 1982).

3.2.3 Lower Concentration Ratios

As we noted in Sec. 3.1, the design of a parabolic trough receiver is influenced by the operating temperature for which it is designed. Larger absorber-tube diameters (i.e., lower concentration ratios) should be chosen for operations at a temperature of $150^{o}C$ than would be selected for operations near $300^{o}C$. The sensitivity of a collector's performance to concentration ratio was discussed in Sec. 3.1 and is illustrated in Fig. 3-1. The design of the drive system and the rigidity of the concentrator will also affect the choice of concentration ratio. If the component design changes discussed in

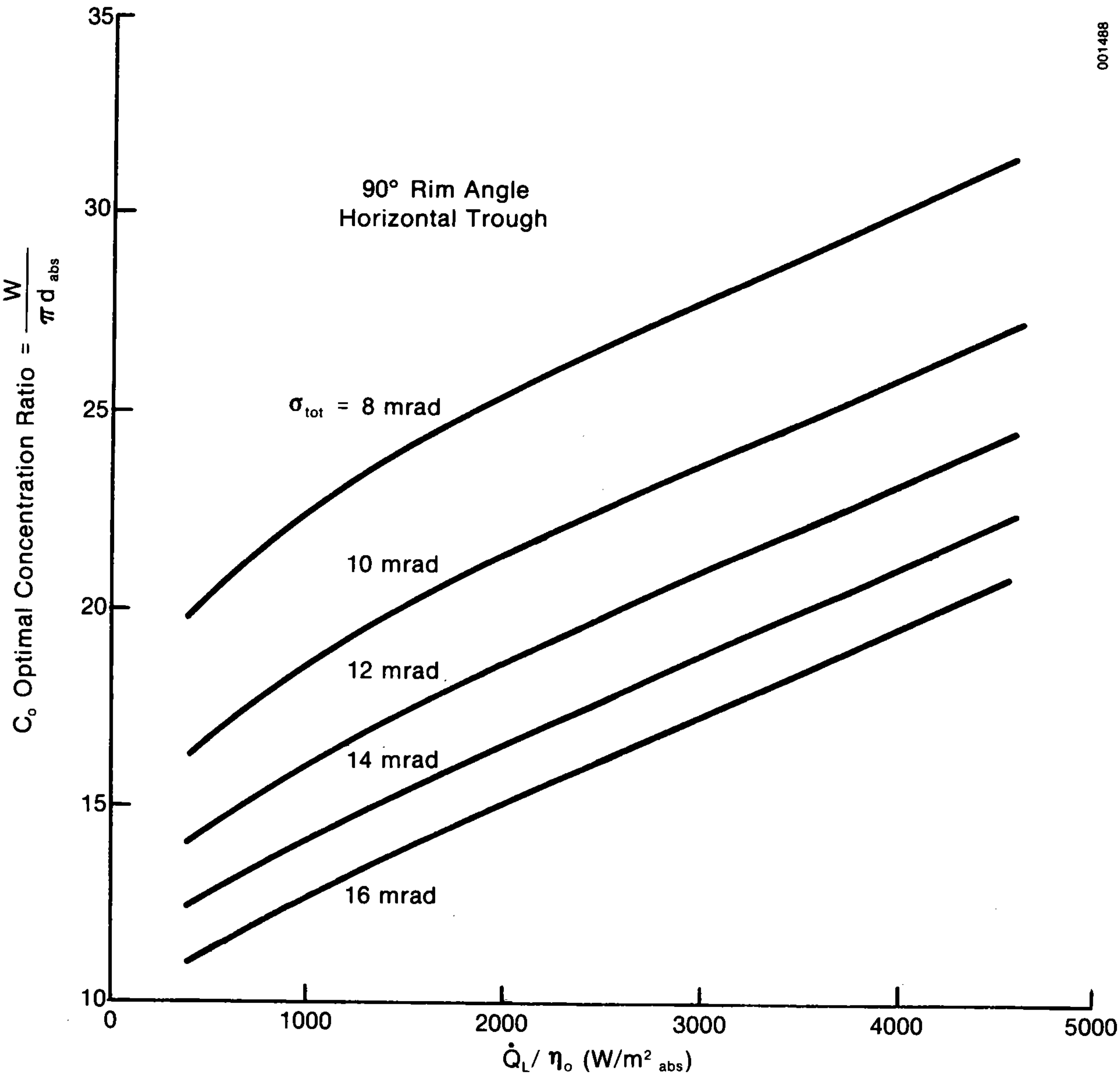


Figure 3-6. Optimum Concentration Ratio vs. Receiver Heat Loss

Sec. 3.2.1 and 3.2.2 are incorporated into the collector's design, the overall optical precision of the collector will be lessened to some degree. This will tend to lower the optimum concentration ratio of the collector further.

The effects of both operating temperature and overall collector optical precision on optimum concentration ratio are quantified in Fig. 3-6, which is based on a concentration ratio optimization procedure specifically developed for parabolic troughs (Rabl et al. 1979). This optimization procedure requires two principal quantities: Q_L/η_o and σ_{tot}.* The first quantity is shown along the abscissa and is the heat-loss rate of the receiver per unit of absorber-tube surface area divided by the optical efficiency of the collector. Values for Q_L may be obtained from Fig. 3-4 for various types of receivers. The second quantity, σ_{tot}, is the total all-day average rms beam spread resulting from all sources of optical error. It is calculated as

$$\sigma_{tot} = [4\sigma^2_{con} + \sigma^2_{spec} + \lambda(\theta)(4\sigma^2_{con} + \sigma^2_{spec}) + \sigma^2_{track} + \sigma^2_{disp} + \sigma^2_{sun}]^{1/2} ,$$

where

σ_{sun} = all-day average value of rms angular width of the sun (as viewed by the receiver)

$\lambda(\theta)$ = coefficient that accounts for the rim-angle-dependent contribution of longitudinal mirror errors to transverse beam spreading

and the other terms are the same as those defined earlier for $\sigma_{optical}$.

Rabl (1979) recommends a value of 0.1 for $\lambda(\theta)$ for concentrators with rim angles between 80^o and 100^o. A value of 5 mrad is also recommended for σ_{sun}, the all-day average effective sun width. Values for the remaining terms depend on the characteristics of the collector components.

Note from Fig. 3-6 that the optimum concentration ratio is quite dependent on the receiver heat-loss rate. For high temperatures (and, therefore, high Q_L values), concentration ratios of 20 to 30 are optimal.** For lower temperature operation, concentration ratios should be significantly lower--in the 10-to-20 range. The value of σ_{tot} further defines the optimum concentration ratio within this range.

At an average operating temperature of 290^oC (550^oF), the heat-loss rate from a typical receiver is about 2660 W/m^2_{abs}. For an assumed optical efficiency of 0.8, the x-axis value of Fig. 3-6 is 3325 W/m^2. A good σ_{tot} value for a very optically precise parabolic trough is 8 mrad. (This is based on σ_{con} = 2.5 mrad, σ_{spec} = 2 mrad, σ_{track} = 2 mrad and σ_{disp} = 2 mrad). For these values, the optimum concentration ratio is shown in Fig. 3-6 to be about 28.

---

*The sensitivity of the concentration ratio optimization to other variables (e.g., geographic latitude, annual irradiation, and receiver shading) is very small, typically resulting in a concentration ratio change of less than one.

**Note that C_o is defined as the optimal geometric concentration ratio, which is calculated as $C_o = W/(\pi d_{abs})$, where W = aperture width.

This is in close agreement with the geometric concentration ratio of 25 recommended by Treadwell (1980) for an optically precise parabolic trough operating at near 290°C (550°F).

For less optically precise parabolic troughs operating at lower temperatures, Fig. 3-6 shows that a significantly lower concentration ratio should be utilized. For example, at 150°C the heat-loss rate from a typical receiver is about 925 W/m^2_{abs}. Again assuming an optical efficiency of 0.8, the x-axis value of Fig. 3-6 is 1155 W/m^2. Choosing a σ_{tot} value of 14 mrad as an estimate of reduced-cost, lower precision trough optics, the optimal concentration ratio is shown to be just over 14. This is just about one-half the concentration ratio recommended for the optically precise, high-temperature trough. Thus, a larger receiver by almost a factor of two is recommended for this lower temperature receiver design.

SECTION 4.0

CONCLUSIONS

Parabolic trough component design is significantly impacted by the operating temperature used as a design point. Commercially available parabolic troughs suitable for solar IPH systems have been generally designed to operate at or near 300°C. At this temperature, collector optical precision is very important. An optical error budget (normal incidence) of 7 mrad has been established for operation at this temperature. Meeting this optical error budget requires very precise parabolic concentrators and accurate sun-tracking and drive systems. However, IPH energy use at below 150°C (300°F) is over five times greater than energy used at from 200°C to 315°C (400°F to 600°F). For this lower temperature range, parabolic trough designs can be significantly different from current designs. The high optical precision requirements established for high-temperature operation can be relaxed significantly in designs for lower temperature operation.

Relaxed optical error budgets can greatly influence the design of parabolic trough components. Relaxed optical requirements allow us to use less rigid, more lightweight concentrators and materials, and constructions like slabs and laminates that used to be considered inappropriate. The trough drive system is similarly affected by designing to a lower temperature. The feasibility of multiple-row drive systems is enhanced in designs for 150°C, because the performance loss from tracking errors can be minimized at low temperatures. Receiver design also depends on design-point temperature. At 150°C, the absorber-tube size could be twice the diameter selected at 300°C.

Together, these low-temperature design opportunities appear to offer significant potential for system cost reductions. Because these design changes represent a major departure from conventional parabolic trough designs, detailed investigation is needed to quantify their cost-saving potential accurately. However, it is clear that lower temperature design will enhance the prospects of developing parabolic trough systems that are cost effective for industrial applications.

SECTION 5.0

REFERENCES

Beckman, W. A.; Klein, S. A.; Duffie, J. A. 1977. Solar Heating Design by the F-Chart Method. New York: John Wiley and Sons.

Bergeron, K. D.; Champion, R. L.; Hunke, R. W. 1980 (Apr.). Line-Focus Solar Thermal Energy Technology Development FY79 Annual Report for Department 4720. SAND 80-0865 (Revised). Albuquerque, NM: Sandia National Laboratories.

Champion, R. L. 1980. "Sheet Molding Compound (SMC) Reflector/Structures." Proceedings of Line-Focus Solar Thermal Energy Technology Development. 9-11 Sept. 1980. Albuquerque, NM: Sandia National Laboratories.

Collares-Pereira, M.; Rabl, A. 1979. "A Simple Procedure for Predicting Long-Term Average Performance of Nonconcentrating and Concentrating Solar Collectors." Solar Energy. Vol. 23; pp. 235-53.

Dickinson, W. C. 1980. "The Role of Financing in the Marketability of Capital-Intensive Solar Technologies for Industry." Proceedings of the 1980 Solar Industrial Process Heat Conference. 16-19 Dec. 1980; Houston, TX.

Edelstein, R. B. 1980. "Cost Goals for Solar Thermal IPH Systems." Solar Industrial Process Heat Conference Proceedings. 16-19 Dec. 1980. Houston, TX.

Edelstein, R.; Flowers, L.; Sharp, J.; Teresawa, K.; Vindum, Y. 1982 (Feb. 26). "Final Presentation of the Solar Thermal Cost Goals Committee." Presented to the U.S. Department of Energy, Washington, D.C.

Flowers, L. 1982 (July). Personal communication, Solar Energy Research Institute.

Gee, R.; Gaul, H. W.; Kearney, D.; Rabl, A. 1980 (Mar.). Long-Term Average Performance Benefits of Parabolic Trough Improvements. SERI/TR-632-439. Golden, CO: Solar Energy Research Institute.

Gee, R.; Murphy, L. M. 1981 (Aug.). Near-Term Improvements in Parabolic Troughs: An Economic and Performance Assessment. SERI/TR-632-870. Golden, CO: Solar Energy Research Institute.

Harrison, T. D. 1981 (Mar.). Midtemperature Solar Systems Test Facility Predictions for Thermal Performance Based on Test Data. SAND 80-1964/7. Albuquerque, NM: Sandia National Laboratories.

Hooker, D. W.; May, E. K.; West, R. E. 1980 (May). Industrial Process Heat Case Studies. SERI/TR-733-323. Golden, CO: Solar Energy Research Institute.

Krawiec, F. 1981 (July). Energy End-Use Requirements in Manufacturing. SERI/TR-733-790R. Golden, CO: Solar Energy Research Institute.

Kutscher, C. F.; Davenport, R. L.; Dougherty, D. A.; Gee, R. C.; Masterson, D. M.; May, E. K. 1982 (Aug.). Design Approaches for Solar Industrial Process Heat Systems. SERI/TR-253-1356. Golden, CO: Solar Energy Research Institute.

Kutscher, C. F.; Davenport, R. L. 1981 (June). Preliminary Operational Results of the Low-Temperature Solar Industrial Process Heat Field Tests. SERI/TR-632-385R. Golden, CO: Solar Energy Research Institute.

Murphy, L. M. 1982 (Aug.). Structural Design Considerations for a Line-Focus Reflective Module Using Inexpensive Composite Materials. SERI/TR-253-1450. Golden, CO: Solar Energy Research Institute.

Murphy, L. M. 1981 (Mar. 25). Optimal Weight Reflective Modules. Solar Energy Research Institute; internal memorandum.

Murphy L. M.; Gee, R. C. 1982 (June). Lightweight Rim-Driven Trough Prototype Experiment. Solar Energy Research Institute; briefing package.

Rabl, A.; Bendt, P.; Reed, K. A.; Gaul, H. W. 1979. Optical Analysis and Optimization of Line-Focus Solar Collectors. SERI/TR-34-092. Golden, CO: Solar Energy Research Institute.

Reuter, R. C. "Initial Parabolic Trough Design and Parametric Evaluation." Proceedings of the Line-Focus Solar Thermal Energy Technology Development Conference. 9-11 Sept. 1980. Albuquerque, NM: Sandia National Laboratories.

Treadwell, G. W.; Grandjean, N. R.; Biggs, F. 1980 (Mar.). An Analysis of the Influence of Geography and Weather on Parabolic Trough Solar Collector Design. SAND 79-2032. Albuquerque, NM: Sandia National Laboratories.

Treadwell, G. W.; Grandjean, N. R. 1981 (Apr.). Systematic Rotation and Receiver Location Error Effects on Parabolic Trough Annual Performance. SAND 81-0159. Albuquerque, NM: Sandia National Laboratories.

APPENDIX

ANNUAL PERFORMANCE MODEL OF A PARABOLIC TROUGH

Parabolic trough annual efficiency was shown in Figs. 3-2 and 3-3 to be a function of three independent variable groupings. These are $\rho\tau\alpha$, Q_L, and $\sigma_{optical}$. This appendix describes how these variable groupings were used to compute η_{coll}, the annual collection efficiency.

Annual collection efficiency is defined as

$$\bar{\eta}_{coll} = \frac{Q_{coll}}{H_{coll}} , \tag{A-1}$$

where

Q_{coll} = annual average energy collection

H_{coll} = annual average irradiation incident on collector aperture.

A utilizability method (Collares-Pereira and Rabl 1979) was used to evaluate Eq. A-1. The values of Q_{coll} and H_{coll} are calculated as the sum of 12 monthly values of energy collection and incident irradiation denoted as $Q_{coll,m}$ and $H_{coll,m}$. These monthly values are based on long-term average irradiation and energy collection for the central day of each month. The long-term average irradiation for the central day of the month $H_{coll,d}$ is calculated by integrating the product of the beam insolation and the cosine of the incidence angle on the collector in the following way:

$$H_{coll,d} = \frac{2t_c}{\omega_c} \int_0^{\omega_c} I_b \cos\theta \, d\omega . \tag{A-2}$$

where

t_c = average operating time of collector after solar noon

ω_c = average startup (or shutdown) hour angle

I_b = direct-normal irradiance

θ = incidence angle of irradiance.

The variation of direct-normal irradiance during an average day depends, to an excellent approximation, on only the length of the day (Zt_c), the time of year, and the site-specific clearness index. The equations used to compute direct-normal irradiance are given by Collares-Pereira and Rabl (1979).

The numerator of Eq. A-1 is the sum of twelve monthly values of average energy collection. Long-term average monthly energy collection is evaluated as

$$Q_{coll,m} = N\, H_{coll,d} \langle \eta_o \rangle \phi \tag{A-3}$$

where

N = number of days in the month

$H_{coll,d}$ = irradiation incident on collector during average day

$\langle\eta_o\rangle$ = day-long average optical efficiency

ϕ = utilizability factor.

$H_{coll,d}$ is calculated as shown earlier in Eq. A-2.

The calculation of $\langle\eta_o\rangle$ requires weighting the product of the optical efficiency at 0^o incidence, η_o, and the incidence angle modifier, $K(\theta)$, to the available beam insolation $I_b \cos\theta$ in the following fashion:

$$\langle\eta_o\rangle = \frac{\int_0^{\omega_c} [I_b \cos\theta]\ \eta_o K(\theta) d\omega}{\int_0^{\omega_c} I_b \cos\theta\ d\omega} . \tag{A-4}$$

The incidence angle modifier $K(\theta)$ defines how the optical efficiency decreases with incidence angle relative to the trough's normal incidence optical efficiency. Several factors contribute to the decrease of optical efficiency with increasing incidence angle. These factors include, in part, the angular dependence of glass annulus transmittance and receiver absorptance. Also, the intercept factor (defined as that fraction of rays incident upon the aperture that reach the receiver) decreases with incidence angle. This decrease in optical efficiency by the intercept factor is brought about in two ways. First, there is beam spreading because of contour and nonspecularity errors. Second, the apparent sun image becomes wider because of the longer reflected path length.

Calculation of the intercept factor γ involves the convolution of the geometric angular acceptance function for a parabolic trough with a Gaussian distribution that accounts for total beam spreading (i.e., both optics and sun size). The intercept factor γ can be expressed as a function of the product of σ_{total} and C. C, the geometric concentration ratio of the trough, is defined as

$$C = \frac{W}{\pi d_{abs}} .$$

σ_{total} is the total rms angular beam spread of the reflected beam from the concentrator to the receiver. It is calculated as

$$\sigma_{total} = \left(\sigma^2_{optical} + \sigma^2_{sun}\right)^{1/2} , \tag{A-5}$$

where

$$\sigma_{optical} = 4\sigma^2_{con} + 0.215 \tan^2\theta\left(4\sigma^2_{con} + \sigma^2_{spec}\right)$$

$$+ \sigma^2_{track} + \sigma^2_{spec} + \sigma^2_{disp}$$

$$\sigma_{sun} = \sigma_{sun,\ noon}/\cos^2(\theta) \ .$$

The sun's rms angular width typically is taken as 2.8 mrad.

After both C and σ_{total} are defined, the intercept factor can be calculated for a 90° rim angle trough with the following equation [curve-fit to Fig. 4-1 of Rabl et al. (1979)],

for $\sigma = \sigma_{total}$,

$$\begin{aligned} \gamma &= 1, \text{ for } \sigma C \leq 0.134 \ ; \\ &= [0.932 + 1.27\ \sigma C - 6.54(\sigma C)^2 + 5.91(\sigma C)^3] \\ &\quad \text{for } 0.1314 < \sigma C \leq 0.45 \ ; \\ &= [1.38 - 2.01\ \sigma C + 1.35(\sigma C)^2 - 0.348(\sigma C)^3] \\ &\quad \text{for } \sigma C > 0.45 \ . \end{aligned} \tag{A-6}$$

The final term in Eq. A-3 is the utilizability factor. The utilizability factor ϕ is defined to account for both the daily heat loss and the variability of the weather. The utilizability factor for a concentrating collector (Collares-Pereira and Rabl 1979) is a function of clearness index K_h and critical intensity ratio X:

$$\phi = 1 - \left(0.049 + 1.44K_h\right)X + 0.341K_hX^2 \ . \tag{A-7}$$

The clearness index K_h is a site-specific parameter. Monthly values for K_h are available for a large number of locations (Beckman, Klein, and Duffie 1977).

The critical intensity ratio X is the ratio of the daily loss to that fraction of the incident solar energy received by the absorber. This ratio is given as

$$X = \frac{U_L\left(T_{abs} - T_{amb}\right)2t_c}{\langle\eta_o\rangle C\ H_{coll}} \ . \tag{A-8}$$

The heat-loss coefficient U_L is a function only of the receiver characteristics. Note that it is defined in terms of absorber tube surface area. A thermal model of a parabolic trough receiver is used to calculate the heat-loss coefficients. The one-dimensional (radial) model assumes that the receiver is in steady state and that the absorber tube and glass envelope are very long concentric cylinders. The heat-loss rate from the absorber tube, $\dot{Q}_L$, is found by the iterative solution of these equations:

$$\begin{aligned} \dot{Q}_L = \dot{Q}_{abs,rad} + \dot{Q}_{abs,cond/conv} &= \dot{Q}_{glass,cond} \\ \dot{Q}_{glass,abs} + \dot{Q}_{glass,cond} &= \dot{Q}_{envir,conv} + \dot{Q}_{envir,rad} \ , \end{aligned} \tag{A-9}$$

where

$\dot{Q}_{abs,rad}$ = radiative heat-loss rate from absorber tube to glazing

$\dot{Q}_{abs,cond/conv}$ = conductive/convective heat-loss rate from absorber tube to glazing

$\dot{Q}_{glass,cond}$ = conductive heat-loss rate through glazing

$\dot{Q}_{glass,abs}$ = rate of heat input into receiver glazing due to solar absorptance of glass

$\dot{Q}_{envir,conv}$ = convective heat-loss rate from glazing to environment

$\dot{Q}_{envir,rad}$ = radiative heat-loss rate from glazing to environment.

Each heat-transfer-rate term is described in detail below per unit length of receiver:

$$\frac{\dot{Q}_{abs,rad}}{\ell} = \frac{\pi d_{abs}\,\sigma\left(T_{abs}^4 - T_{glass,i}^4\right)}{\frac{1}{\varepsilon_{abs}} + \frac{d_{abs}}{d_{glass,i}} \quad \frac{1}{\varepsilon_{glass}} - 1} \tag{A-10}$$

where

d_{abs} = absorber tube outside diameter

$d_{glass,i}$ = inside diameter of receiver glazing

T_{abs} = absorber tube surface temperature

$T_{glass,i}$ = temperature of receiver glazing inside surface

ε_{abs} = absorber tube emittance

ε_{glass} = emittance of receiver glazing

σ = Stephan-Boltzman constant.

$$\frac{\dot{Q}_{abs,cond/conv}}{\ell} = \overline{Nu}\ \pi K_{gas}\left(T_{abs} - T_{glass}\right) , \tag{A-11}$$

where

$\overline{Nu}$ = combined conductive/convective Nusselt number

K_{gas} = thermal conductivity of receiver annulus gas .

$$\frac{\dot{Q}_{glass,cond}}{\ell} = \frac{2\pi K_{glass}}{\ln \frac{d_{glass,o}}{d_{glass,i}}} \left(T_{glass,i} - T_{glass,o}\right) , \tag{A-12}$$

where

K_{glass} = thermal conductivity of receiver glazing

$T_{glass,o}$ = temperature of receiver glazing outside surface

$d_{glass,o}$ = outside diameter of receiver glazing.

$$\frac{\dot{Q}_{glass,abs}}{\ell} = \alpha_{glass} I_b \rho W , \tag{A-13}$$

where

α_{glass} = receiver glazing absorptance

ρ = concentrator reflectance

W = concentrator aperture width.

$$\frac{\dot{Q}_{envir,conv}}{\ell} = \pi K_{air} Nu_{wind} \left(T_{glass,o} - T_{amb}\right) , \tag{A-14}$$

where

Nu_{wind} = wind-velocity-induced Nusselt number .

$$\frac{\dot{Q}_{envir,rad}}{\ell} = \varepsilon_{glass} \pi d_{glass,o} \sigma \left(T^4_{glass,o} - T^4_{sink}\right) , \tag{A-15}$$

where

T_{sink} = radiative sink temperature of environment (average of ambient and effective sky temperatures).

The properties of air, glass, annulus gas, and black chrome are computed as a function of their temperature. The term $Q_{glass,abs}$ is a function of beam insolation and, therefore, varies during the day. However, the sensitivity of the heat-loss coefficient to this term is small and is, therefore, assumed constant to simplify the results. A wind velocity of 2 m/s (4.5 mph) over the receiver is also assumed as representative of average wind conditions.

Receiver heat loss can be expressed as a heat-loss coefficient, U_L, that is based on absorber tube surface area. U_L is related to receiver heat loss by the following equation:

$$U_L = \frac{\dot{Q}_L}{\pi d_{abs} \ \ell \left(T_{abs} - T_{amb}\right)} . \tag{A-16}$$

Figure 3-4 shows the solution of the heat rate equations, in terms of U_L, for several types of parabolic trough receivers. A 2.54-cm absorber-tube diameter and an ambient temperature of 10^oC have been assumed. However, the variation of U_L with absorber tube diameter and ambient temperature has been shown to be small (Gee et al. 1980).

Document Control Page	1. SERI Report No. SERI/TR-253-1662	2. NTIS Accession No.	3. Recipient's Accession No.
4. Title and Subtitle Low-Temperature IPH Parabolic Troughs: Design Variations and Cost-Reduction Potential			5. Publication Date November 1983 6.
7. Author(s) Randy C. Gee			8. Performing Organization Rept. No.
9. Performing Organization Name and Address Solar Energy Research Institute 1617 Cole Boulevard Golden, Colorado 80401			10. Project/Task/Work Unit No. 1384.30 11. Contract (C) or Grant (G) No. (C) (G)
12. Sponsoring Organization Name and Address			13. Type of Report & Period Covered Technical Report 14.
15. Supplementary Notes			

16. Abstract (Limit: 200 words)

Parabolic trough concentrating collectors have improved in performance and reliability since the U.S. Department of Energy's solar thermal program began. The development of components has been tailored to meeting instantaneous collector performance goals at an operating temperature of 315°C (600°F). Although this strategy has resulted in advanced component designs and excellent collector performance at 315°C, it has also necessitated somewhat stringent, expensive component design requirements. Near-term parabolic trough collector systems are too expensive to compete widely with conventional fossil-fuel systems. This report describes how collector costs can be reduced with lower-temperature designs. Industrial process heating energy use at below 150°C (300°F) is over five times greater than that at from 200° to 315°C (400°F to 600°F). Component designs for lower temperatures are considered, including less rigid, lighter-weight concentrators; alternative concentrator constructions; larger receivers; and multiple-row drive systems.

17. Document Analysis

a. Descriptors Cost ; Design ; Economics ; Efficiency ; Industry ; Parabolic Collectors ; Parabolic Trough Collectors ; Performance ; Process heat ; Solar Collectors

b. Identifiers/Open-Ended Terms Industrial Process Heat

c. UC Categories

59b,59c,62a

18. Availability Statement	19. No. of Pages
National Technical Information Service U.S. Department of Commerce 2585 Port Royal Road Springfield, Virginia 22161	42 20. Price A03

Form No. 0069 (3-25-82)

UNITED STATES DEPARTMENT OF ENERGY
P.O. BOX 62
OAK RIDGE, TENNESSEE 37830

OFFICIAL BUSINESS
PENALTY FOR PRIVATE USE, $300

POSTAGE AND FEES PAID
UNITED STATES
DEPARTMENT OF ENERGY

U S MAIL

FS- 1

10723
US DEPARTMENT OF ENERGY
ATTN S D ELLIOTT, DIR
SOLAR ONE PROJECT OFFICE
PO BOX 366
DAGGETT, CA 92327