DESIGN AND TEST OF NON-EVACUATED SOLAR COLLECTORS WITH COMPOUND PARABOLIC CONCENTRATORS

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Abstract

The intermediate range of concentration ratios (1.5X-10X) which can be achieved with CPC's without diurnal tracking provides both economic and thermal advantages for solar collector design even when used with non-evacuated absorbers. The present paper summarizes more than 3 years of research on non-evacuated CPC's and reviews measured performance data and critical design considerations. Concentrations in the upper portions of the practical range (e.g. 6X) can provide good efficiency (40% to 50%) in the 100°C - 160°C temperature range with relatively frequent tilt adjustments (12-20 times per year). At lower concentrations (e.g. 3X) performance will still be substantially better than that for a double glazed flat plate collector above about 70°C and competitive below, while requiring only semi-annual adjustments for year round operation. In both cases the cost savings associated with inexpensive reflectors, and the optimal coupling to smaller, simple inexpensive absorbers (e.g. tubes, fins, etc.) can be as important an advantage as the improved thermal performance.

The design problems for non-evacuated CPC collectors are entirely different from those for CPC collectors with evacuated receivers. For example, heat loss through the reflector can become critical, since ideal CPC optics demands that the reflector extend all the way to the absorber. Recent improvements in reflector surfaces and low cost antireflection coatings have made practical a double-glazed non-evacuated CPC design. It is calculated that a 1.5X version of such a collector would have an optical efficiency $\eta_o = 0.71$, a heat loss coefficient $U = 2.2$ W/m^2°C and a heat extraction efficiency factor $F' \geq 0.98$, while requiring no tilt adjustments.

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I. Introduction

In 1974 when the suitability of the Compound Parabolic Concentrator (CPC) for solar energy collection was recognized in the U.S.,[1] a research program was begun to build and test collectors of this type. Such collectors attain or closely approach the maximum concentration possible for a given acceptance angle (field of view) making possible intermediate concentration levels with only seasonal tilt adjustments. For the first generation of CPC collectors[2] the optical performance and the convective and radiative heat losses agreed with those predicted. However, the efficiency at high temperature was somewhat poorer than expected. Careful analysis of the data showed that this was due to conductive losses through the reflector and/or the insulation. This experience serves to emphasize the care which must be taken when implementing this novel design principle in solar energy applications.

One solution to this problem presented itself when evacuated receiver tubes[3] of potentially low cost became available in 1975. By coupling CPC reflectors with evacuated receiver tubes efficiencies above fifty percent in the temperature range of 100 to 200°C have been demonstrated with non-tracking solar collectors[4], and even 300°C at reasonable efficiency appears to be feasible for fixed collectors of advanced design[5]. On the other hand at lower temperatures, around 100°C, calculations[6] indicated that even non-evacuated CPC collectors with proper design can operate with acceptable efficiency, surpassing available flat plate collectors. The present paper describes the design, construction and test of two prototype non-evacuated CPC collectors with concentration ratios of 6.5 and 3.0 and analyzes the test results. After completion of the first phase of testing, the initial version[7] of the 6.5X collector was modified to have enlarged absorber tubes to increase its optical tolerances. The use of oversized absorber tubes reduced the net concentration to 5.2 hence this version of the collector, which exhibited the best performance achieved, is referred to as the 5.2X. This performance is illustrated in Figure 1, where the measured efficiency of the 5.2X is superimposed on a performance prediction which was published several years ago[6]. The data can be characterized by an optical efficiency $\eta_o = 0.68$ and a heat loss coefficient, $U = 1.85\ W/m^2\,°C$. The agreement between this observed performance and the predictions confirms that CPC collectors of a design suitable for practical application can be reliably designed and built. It took a slow and sometimes painful learning experience to reach this point, and for the benefit of future efforts in development or manufacture of CPC collectors we summarize the crucial design considerations. Details can be found in References 2, 6 and 8.

CPC reflectors can be designed for any absorber shape: for example, flat one-sided absorbers as in Figure 2a, flat two-sided absorbers (fins) as in Figure 2b, wedge-like absorbers as in Figure 2c or tubular absorbers as in Figure 2d. The first generation of CPC collectors[2] was based on configuration 2a), and a cavity was used as absorber because high absorptance was thought to outweigh heat losses. Later, analysis of the test data, heat transfer calculations and the emergence of a viable selective coating industry convinced us that the absorber of a CPC should have a selective coating and that a cavity is undesirable because of its high heat losses. Parasitic heat losses through the back and through the reflector can be serious. Conduction through the back is minimized by the "backless" configurations of Fig. 2b) and d); the

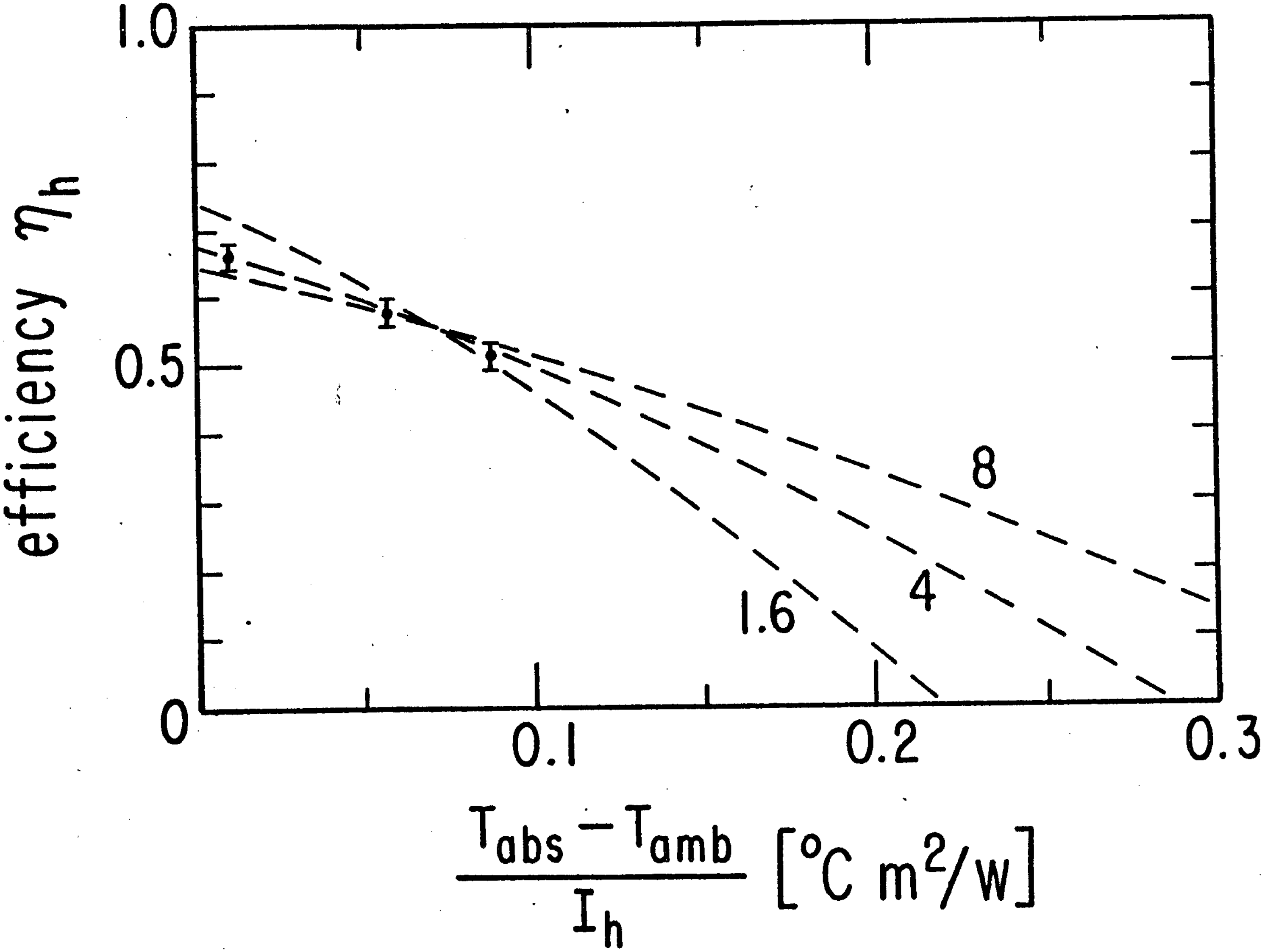

Fig. 1 Efficiency predictions for single glazed non-evacuated CPC with selective coating for concentration values 1.6, 4.0 and 8.0 (from Ref. 6). The data points show the measured performance of a collector of this type with concentration 5.2.

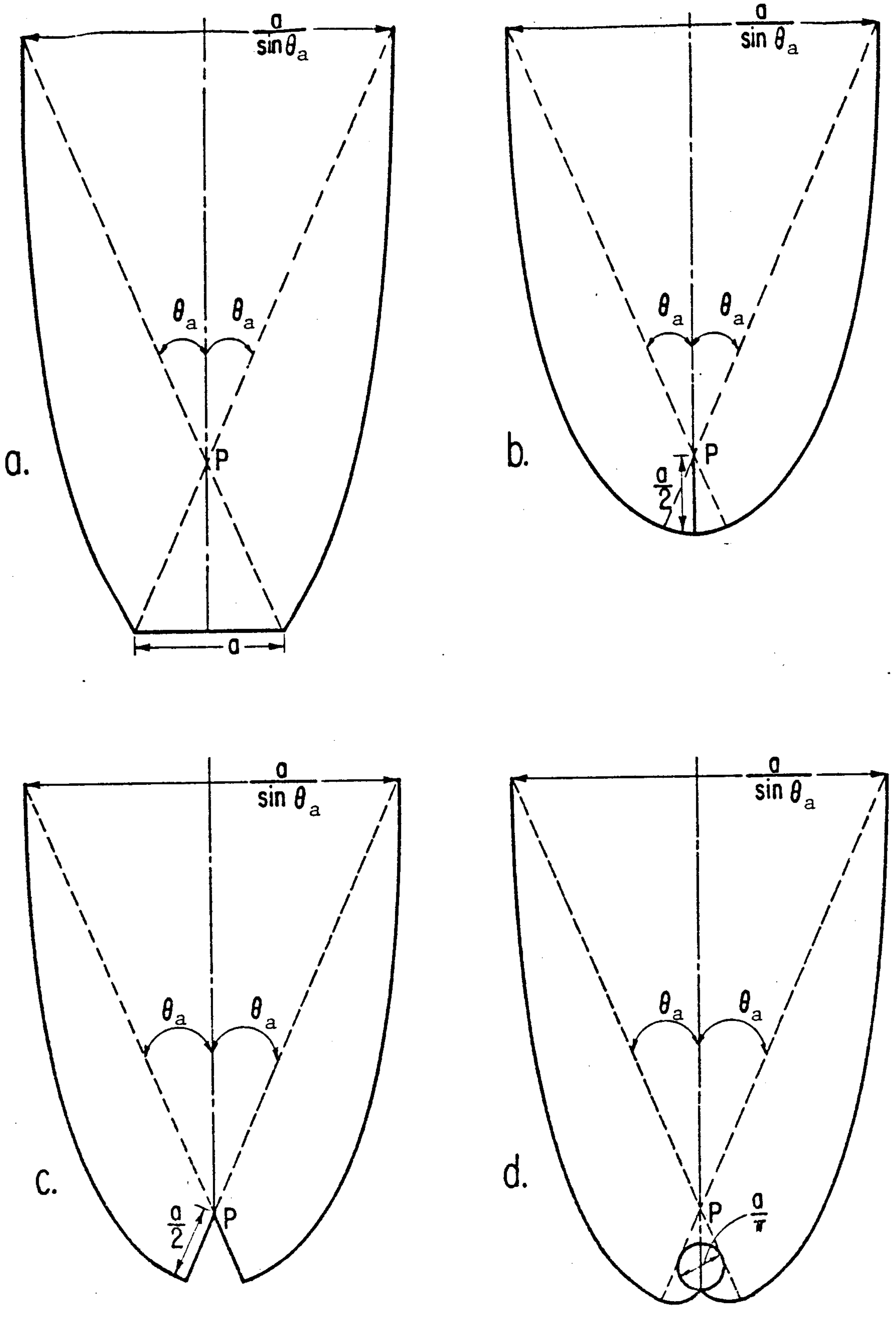

Fig. 2 Four different configurations of the CPC.
a) Flat one-sided receiver
b) Fin receiver
c) Wedge receiver
d) Tubular receiver

heat loss reduction more than compensates for the fact that the average number of reflections is approximately 0.5 higher than for configuration 2a). The "backless" versions are preferable for economic reasons as well because they require only half as much of the relatively expensive absorber material; also, they are less deep and have less reflector surface, as can be seen directly from Fig. 2.

Reduction of heat transfer into the reflector structure is crucial; in fact it is easy to enlarge the heated area so much as to nullify all benefits of concentration. Use of aluminum sheet as reflectors is problematic unless the ratio of absorber width to aluminum thickness is sufficiently large and/or the reflector is sufficiently decoupled from the absorber by a small gap maintained by insulating standoffs or better by means of a glass envelope. In single glazed (cover only) nonevacuated CPC's, aluminized or silvered plastic with foam or fiberglass backing is certainly preferred from a thermal point of view.

Contour accuracy of the reflector surface has never posed a problem with any of the manufacturing techniques that were tried; even an "orange peel" surface turned out to be acceptable. The surface need not be very specular[9] (i.e., look shiny), but its total reflectance should be as high as possible. Correct placement of the absorber relative to the reflector does, however, require some care with the backless configurations. To minimize optical losses and allow for placement tolerances we recommend oversizing of the absorber by about twenty percent, choice of low concentration ratios (less than five), and sufficiently large size (say absorber widths of at least a few centimeters).

The two collector modules which are the basis for the experimental results in this paper were built with different objectives in mind. The first design (the 6.5X) is an experimental collector whose design parameters (acceptance

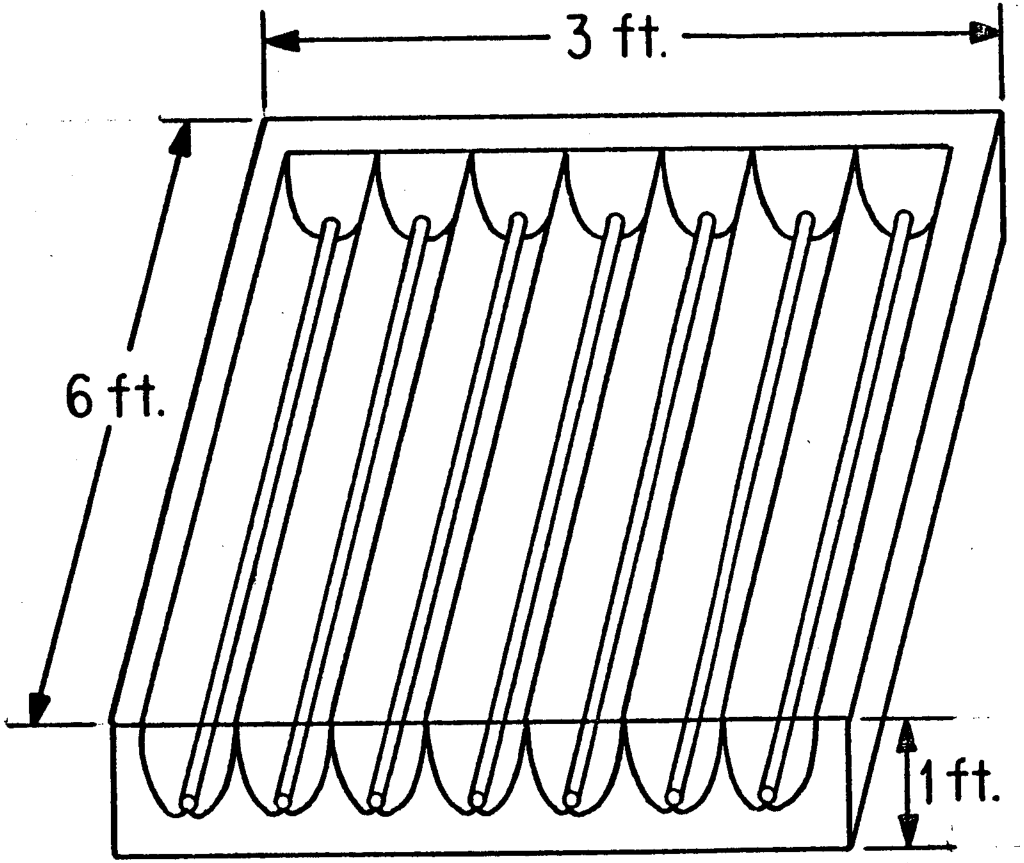

Fig. 3a 6.5X collector panel.

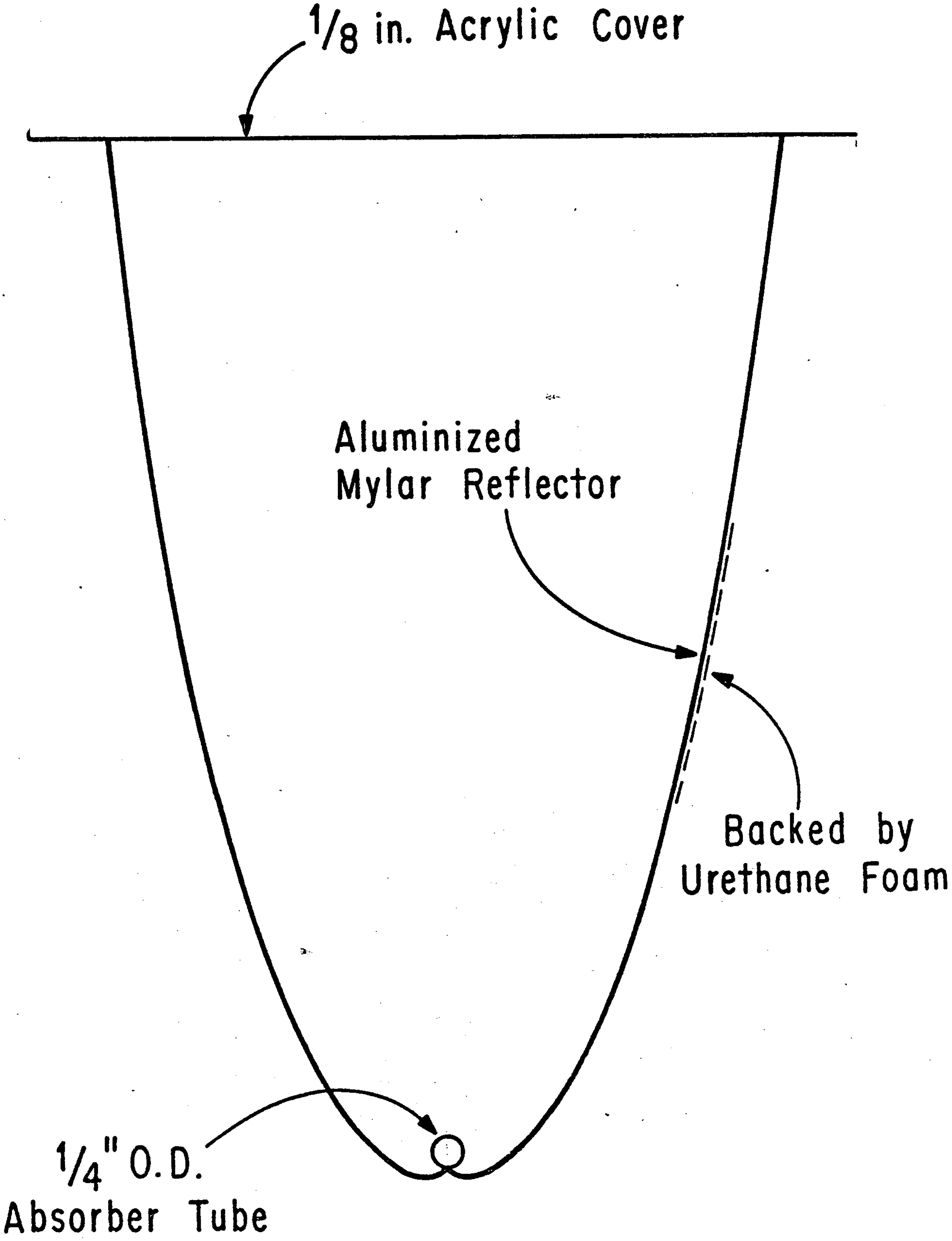

Fig. 3b Cross section of 6.5X.

angle, concentration ratio, degree of truncation, etc.) were selected to examine what was felt to be the limits of tolerance on a practical non-evacuated design. The second (the 3X) was a working prototype of a production (although not commercial) design for the first experimental array of CPC collectors to be installed in a heating application (the Bread Springs Elementary School, Navajo Reservation near Gallup, N.M.)[10]. Both collectors used "backless" configurations.

The 6.5X was built as a panel with 7 CPC troughs and an active area of 91 x 183 cm^2 (3 x 6 ft^2) and depth 30.5 cm (1 ft), as shown in Figure 3a. The absorber was originally a tube with 0.64 cm (1/4") outer diameter, all troughs being connected in series flow. The cross section of the reflector and receiver is in Figure 3b. The design acceptance half angle is 6.4°*, corresponding to an ideal concentration of 9.0. The reflector was truncated to about one-third of its full height, resulting in an actual concentration of 6.5. In the later version described below an oversized absorber of diameter 0.80 cm was used resulting in an effective geometric concentration ratio of 5.2. In order to minimize conductive heat losses through the reflector, the reflectors were fabricated by pouring a high temperature urethane foam over aluminized mylar which had been stretched over a mold with the CPC profile. This method was most convenient for the fabrication of a single research prototype; for actual mass production, different techniques might be more practical (for example, the use of fiberglass plus epoxy).

Since the 3X collector was designed for heating applications only, the design acceptance angle was chosen to be ±18° to insure collection for at least 7 hours a day for the six month period between the fall and spring

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*This requires approximately 24 tilt adjustments per year and allows at least a week between successive adjustments assuming a minimum of seven hours of collection per day.

equinoxes without any tilt adjustments. The absorber is a fin (configuration of Fig. 2b) with center tube, and each collector consists of a two trough module so that the cross section appears as shown in Figure 4. Each module is 76 x 170 cm (14 ft^2) in net area and the troughs were 46 cm (18 in) deep. Kinglux sheet 0.5 mm (20 mils) thick is used as the reflector (total reflectance $\rho = 0.84$) backed by urethane foam in the prototype. In this configuration the large ratio of collector depth to aluminum sheet thickness is sufficient so that the relative heat losses through the reflector are small. The outer enclosure is a fiberglass-epoxy tub with water white glass as a cover. The absorber is a 1.59 cm (5/8 in) outer diameter copper tube with two fins of 2.54 cm (1 in) height plated with black chrome.

The paper is organized as follows: Section II discusses collector parameters and test procedures with emphasis on the special features of the CPC. Preliminary and diagnostic tests are described which have proved valuable for the development of CPC collectors. Since there was, and still is, no standard test procedure for collectors of the CPC type, our tests evolved as the work progressed. Therefore some of the early tests reported in this paper do not conform with what we now recommend as standard test procedure. The specification of certain collector parameters is a matter of convention, for example, the choice of insolation measurement (pyranometer, pyrheliometer or other) and the choice of collector temperature (fluid inlet, mean fluid or other). Lacking a generally accepted test procedure, we have taken care to report all test results with sufficiently detailed information to permit conversion to any other reasonable set of conventions. In Section III design and test results for the 6.5X and for the 5.2X are presented. Section IV deals with the 3X collector. Results are summarized in Section V, and a new design is described,

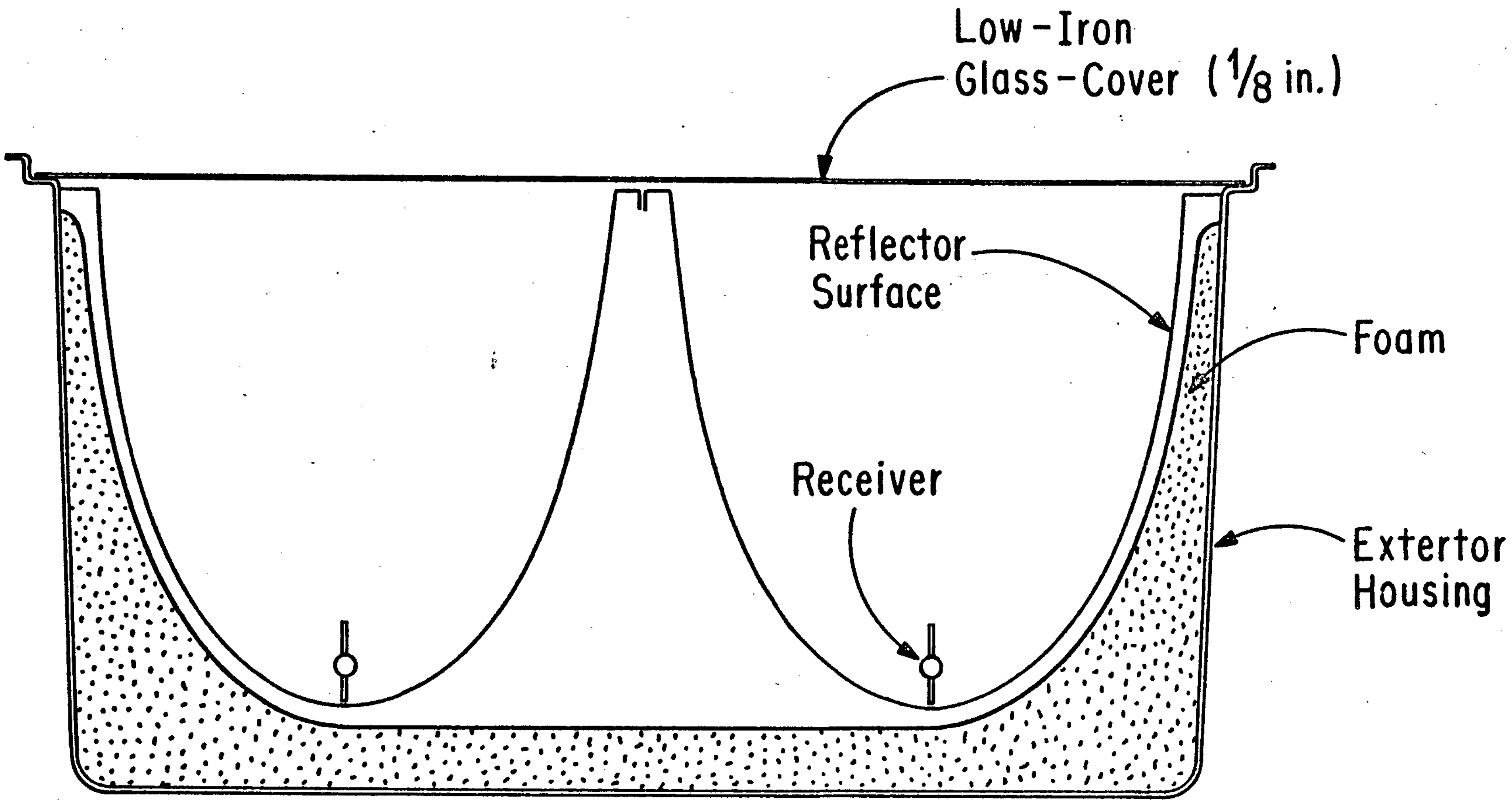

Fig. 4 Cross section of 3X.

a nonevacuated double glazed 1.5X which does not require any tilt adjustments; this collector is preferable to the 5.2X when low cost antireflection coatings are available.

II. Test Procedures

A. Diagnostic Tests

When building a new collector type, it is advisable to avoid expensive unpleasant surprises by first performing certain preliminary tests on a small collector module which is easily fabricated. For the thermal tests the smallness of such a module rules out measurements with heat transfer fluid; also one must construct this module with sufficient back and side insulation to simulate the environment in a complete collector. We briefly describe three tests which are suitable; specific results are reported in the respective sections on the 3X and the 6.5X.

i) Optical Measurements

For a preliminary evaluation of the optical quality of a CPC reflector we found the following techniques most convenient. Visual inspection from various angles gives a measure of the acceptance angle. One simply looks at the aperture of a CPC, with receiver in place, and estimates what fraction of the aperture looks black when viewed from various incidence angles. Ideally, i.e., for perfect reflector contour and for perfect placement of the receiver, the aperture will appear completely black when viewed within the acceptance angle; at the acceptance angle, $|\theta| = \theta_a$, there is a sharp transition from full acceptance to rejection of all rays (for an untruncated CPC) or rejection of most rays (for a truncated CPC). In a real CPC the transition around $|\theta| = \theta_a$ is smeared out over an angular range which is approximately four times the average contour error of the CPC. Such visual estimate of the fraction of the aperture which appears black, is accurate enough despite its subjective element to determine the useful acceptance angle of a collector within a fraction of degree. The only cautionary note about this test concerns the distance from which the aperture must be viewed: for a finite viewing distance the angle from the eye to the left edge of the aperture differs from the angle to the right edge, and this difference limits the accuracy to which the acceptance angle can be determined. One should be a distance L away which is large (i.e., $\gtrsim$ 50 times) compared to the aperture. This test also reveals losses due to contour errors or receiver misplacement if the aperture shows significant reflective spots when viewed well within the acceptance angle.

Optical measurements of the cover transmittance τ_{cover}, of the reflectance ρ of the CPC wall and of the absorptance α of the receiver are necessary. Knowing ρ one can approximate the throughput or effective transmittance of the CPC by the formula

$$\tau_{CPC} = \rho^{<n>} \qquad \text{(II-1)}$$

where $<n>$ is the average number of reflections which has been calculated in Refs. 6, 15 and 16. For the CPC's in this paper τ_{CPC} can be measured directly based on the following observation: a CPC of the fin or tubular configuration and with a perfect reflector in place of the receiver is optically equivalent to the same CPC without any receiver. Therefore the effective reflectance of the aperture of a CPC without receiver is $(\tau_{CPC})^2$. The usual spectrophotometers are too small to measure this reflectance. Instead we built a large reflectometer, called "light box", consisting of a phototube as detector and a cubical box, 70 cm each side, with fluorescent light tubes and white walls on the inside as an integrating "spherical" source. Having measured τ_{cover}, α and τ_{CPC} one can predict the optical efficiency with respect to radiation within the acceptance angle as

$$\begin{aligned} \eta_{o,C} &= \tau_{cover}\,\tau_{CPC}\,\alpha \\ &= \tau_{cover}\,\rho^{<n>}\,\alpha \end{aligned} \qquad \text{(II-2)}$$

unless the optical efficiency is further reduced by an intercept factor corresponding to radiation missing the receiver.

ii) Heat-Loss Measurements

The second test series measures the heat loss by letting the absorber reach equilibrium when it is heated by electric resistance heating. This determination of the heat loss coefficient

$$U_{lab} = \frac{q_{electric}}{A\,(T_{absorber} - T_{ambient})} \qquad \text{(II-3)}$$

is simple and accurate. This test procedure can also determine the relative magnitude of front and back losses, if the measurement is repeated when the collector module is covered with insulating material or with a plate which is thermostatically controlled to have the same temperature as the absorber. However, one must keep in mind that U_{lab} may differ somewhat from the real heat loss coefficient U under actual operating conditions when the reflectors are warmed by direct absorption of solar radiation.

iii) Masked Stagnation Tests

By holding a mask of known transmittance τ_m, for example, a perforated metal sheet, in front of the collector module, one can control the amount of solar radiation $\eta_o\,\tau_m\,I_h$ reaching the receiver. When the receiver stagnates, i.e. reaches equilibrium the heat losses are exactly equal to this amount of radiation, hence the stagnation temperature T_s satisfies

$$U\ (T_s - T_{ambient}) = \tau_m \eta_{o,h}\ I_h. \qquad \text{(II-4)}$$

This test yields an independent determination of the ratio $U/\eta_{o,h}$ of U-value to optical efficiency.

B. Direct Measurement of Operating Efficiency

The thermal output q [in W] of each collector was determined by measuring (with platinum resistance thermometers) the temperature rise of water* flowing through the collector, and multiplying it by heat capacity c_p and mass flow rate $\dot{m}$

$$q = \dot{m}\ c_p\ (T_{out} - T_{in}). \qquad \text{(II-5)}$$

The flow rate was found by means of stop watch and graduated beaker. For the measurements near ambient temperature, which were taken to determine the behavior of the optical efficiency, we found open loop operation the most convenient, i.e., discarding the water after it has gone once through the collector. With open loop operation it is easy to maintain stable low temperatures for indefinite periods. To relate heat output to efficiency

$$\eta_h = \frac{q}{A\ I_h}, \qquad \text{(II-6)}$$

the hemispherical (also called total) irradiance I_h was measured with a

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*Use of other liquids, in particular commercial anti-freeze, is likely to be inaccurate because the heat capacity may not be known correctly, or it may change during the testing if the liquid changes at high temperature, by decomposition or by evaporation of some component.

pyranometer, mounted in the plane of the collector. The pyranometer* was calibrated for tilt dependence of sensitivity.A is the (net) aperture area of the collector. The fraction γ of the hemispherical irradiance I_h which falls within the acceptance angle of a CPC collector depends on its concentration and on the haziness of the atmosphere, ranging from about 92% for a 3X (90% for a 6.5X) on clear days to 80% or less on hazy days (one should note that for high concentration focusing collectors the corresponding values of γ are significantly lower). Therefore, the collector performance can display considerable scatter when analyzed in terms of hemispherical irradiance I_h under different weather conditions. To minimize this scatter we have tested solar collectors only under reasonably clear sky conditions. During the test series we evolved a simple procedure for monitoring the ratio I_d/I_h of diffuse over hemispherical insolation, for future reference. We measured this ratio I_d/I_h simply by holding an occulting disk in front of the pyranometer to block out the beam radiation. A disk of about 0.1 m diameter held approximately 1 m above the pyranometer is adequate. To provide a correction procedure which compensates for variations in the I_d/I_h ratio we note that the insolation within the acceptance angle of a CPC of geometric concentration C is very well approximated by

$$\begin{aligned} I_C &= I_b + \frac{1}{C} I_d \\ &= I_h + (\frac{1}{C} - 1)\ I_d \end{aligned} \qquad \text{(II-7)}$$

where I_b is the beam (also called direct) component of insolation.

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*Model Eppley 8-48, which had been calibrated against a precision Eppley PSP pyranometer.

The efficiency should therefore be nearly independent of I_d/I_h if it is referred to I_C instead of I_h. This suggests that the efficiency η_h with with respect to hemispherical irradiance be corrected according to

$$\eta_{h,\text{standard}} = \frac{1 + (\frac{1}{C} - 1)\left.\frac{I_d}{I_h}\right|_{h,\text{standard}}}{1 + (\frac{1}{C} - 1)\,\frac{I_d}{I_h}}\;\eta_h. \qquad \text{(II-8)}$$

We have used this formula with a clear day ratio of

$$\left.\frac{I_d}{I_h}\right|_{h,\text{standard}} = 0.11 \qquad \text{(II-9)}$$

to reduce the scatter in the 5.2X data.

Since the insolation within the acceptance angle of a CPC of concentration greater than 2 is closer to the beam irradiance I_b than to the hemispherical irradiance I_h, it is perhaps more appropriate to refer the efficiency to a pyrheliometer. However, since nonevacuated CPC's are low to intermediate temperature collectors, more likely to be compared to flat plates, we maintained the convention of referring their efficiencies to the hemispherical irradiance as indicated by the subscript h of the efficiency.

C. Collector Performance Parameters

It is desirable to summarize collector test results in terms of a few simple parameters. In this subsection we list a set of parameters for CPC collectors which is simple, conforms closely with common practice for other collector types and will permit system performance predictions with better than five percent accuracy.

Experience with solar collector tests and performance calculations has shown that to a good approximation the operating efficiency, η_h, can be characterized by a single curve if it is plotted versus the ratio of the collector temperature (relative to ambient) to the insolation level. In most cases one is interested only in a rather narrow range of temperatures where this curve can be approximated by a straight line. One can therefore write

$$\eta_h = F'[\eta_{o,h} - U\,\Delta T/I_h] \qquad \text{(II-10)}$$

where

$$\Delta T = T_f - T_{ambient} \qquad \text{(II-11)}$$

is the difference between mean fluid temperature

$$T_f = (T_{in} + T_{out})/2 \qquad \text{(II-12)}$$

and ambient and the factor F' accounts for temperature differences between fluid and absorber surface. This point is addressed in the following subsection and shown to be of minor importance for collectors in this paper because F' is very close to unity. The subscripts h for efficiency and insolation indicate the choice of hemispherical irradiance as insolation base (the analogous formula can of course be written down with respect to beam irradiance I_b).

Equation II-10 is based on measurement at normal incidence ($\theta = 0$). To predict long term performance one also needs to know the incidence angle modifier which multiplies $\eta_{o,h}$ in Eq. II-10 if θ is not zero. For the CPC there are two principal angular coordinates, $\theta_{||}$ measured along the trough, and $\theta_{\perp}$ measured perpendicular to it. Ideally the efficiency of a CPC trough with acceptance half angle θ_a should be constant for all

$$|\theta_{||}| < \pi/2 \text{ and } |\theta_{\perp}| < \theta_a. \qquad \text{(II-13)}$$

Due to errors in mirror contour and in receiver placement, the useful angular range in $\theta_{\perp}$ of a real CPC will be slightly smaller than the design acceptance angle $2\theta_a$. In this paper the angular behavior is reported as an angular scan, i.e., a measurement of the incidence angle modifier $\kappa(\theta_{||},\theta_{\perp})$ as function of $\theta_{\perp}$ and of $\theta_{||}$. The $\theta_{\perp}$ scan is sufficiently flat at the center to allow characterization by a single number $\theta_{a\ \text{useful}}$, the angular range over which $F'\eta_o$ attains at least ninety percent of its peak value. In the longitudinal direction the efficiency falls off essentially like the well known incidence angle modifier of a flat plate collector and can be summarized by an all day average value $\bar{\kappa}$.

Most calculations of long term energy delivery by a CPC, in particular Ref. 11, will assume as input the zero ΔT efficiency

$$\eta_C\ (\Delta T = 0)$$

with respect to insolation within the acceptance angle. The value $\eta_h(\Delta T=0)$ Eq. II-10 is based on hemispherical insolation and must therefore be converted according to

$$\eta_C(\Delta T=0) = \frac{\eta_h(\Delta T=0)}{1 + (\frac{1}{C} - 1)\frac{I_d}{I_h}} \qquad \text{(II-14)}$$

where I_d/I_h is the ratio of diffuse over hemispherical insolation during the collector tests and C is the geometric concentration ratio.

D. Interpretation of Performance Parameters

While the user of a collector need not worry about the origin of the collector parameters, their optical and thermal interpretation is of concern to collector designers and manufacturers who want to assess the potential for performance improvement. As a first approximation $\eta(\Delta T=0) = F'\eta_o$ represents the optical throughput from aperture to receiver and U represents the collector heat loss. This simple picture is, however, complicated by several effects, in particular in the case of nonevacuated concentrating collectors.

First of all there is the difference between the absorber surface temperature and the fluid reference temperature. This difference can be accounted for by the heat extraction or heat removal factors of the Hottel-Whillier-Bliss model[12,13], which is well known from the flat plate literature and which is applicable for concentrating collectors as well. If, as in this paper, the mean fluid temperature Eq. II-12 has been used as reference, the relevant factor is the efficiency factor F' for the heat extraction from absorber surface to fluid; it is given by the ratio[12]

$$F' = \frac{R_{ra}}{R_{fr} + R_{ra}} \qquad \text{(II-15)}$$

of the thermal resistance R_{ra} from receiver surface to ambient over the thermal resistance $(R_{fr} + R_{ra})$ from fluid to ambient. R_{ra} is, of course, related to U-value U and aperture area A by

$$R_{ra} = \frac{1}{AU} \qquad \text{(II-16)}$$

The resistance between fluid and receiver surface is given by

$$R_{fr} = \frac{1}{A_r U_{fr}} \qquad \text{(II-17)}$$

where U_{fr} is the conductance from fluid to receiver surface, per receiver surface area A_r. In most tubular receivers of solar collectors the resistance across the tube wall is small compared to the resistance across the fluid film; one can therefore set U_{fr} equal to the heat transfer coefficient of the fluid[14]. When water is used as heat transfer fluid this coefficient is so large, on the order of 1000 W/m^2°C, that the resulting F' is very close to unity and flow rate dependent variations in U_{fr} have no noticeable effect on the collector efficiency. For example, the 6.5X in its revised version has concentration ratio $C = A/A_r = 5.2$ and U-value $U = 1.85$ W/m^2°C, and

$$F' = \frac{1}{1 + C\ U/U_{fr}}$$

$$= \frac{1}{1 + 5.2 \times 1.85/1000} = 0.99 \qquad \text{(II-18)}$$

Since the U-value scales approximately like 1/C, F' is nearly independent of concentration ratio. For the 3X collector we estimate F' to be slightly smaller, about 0.98, because of the additional resistance in the fin.

The next comment concerns the difference between the efficiency at zero ΔT and the optical efficiency η_o, the latter being defined as fraction of available irradiation at the aperture which reaches the absorber and is absorbed. In non-evacuated CPC collectors $\eta_h(\Delta T{=}0)$ is noticeably larger than $\eta_{o,h}$ because direct absorption of solar radiation by the reflector reduces the heat loss from the receiver. This effect is difficult to calculate accurately because it involves close coupling between radiative, convective and conductive heat transfer modes in a relatively complicated geometry. The simple model of Ref. 6 showed that this effect tends to shift the entire efficiency curve upward by a few percentage points. This is plausible because to a first approximation the solar radiation absorbed by the reflector raises the effective average temperature of the environment of the absorber tube by an amount proportional to insolation but nearly independent of collector operating temperature. Relative to a calculation which discards the power absorbed by the reflector, warming of the reflector has therefore the same effect as a reduction in the difference $T_{absorber} - T_{ambient}$. This implies that the entire efficiency curve is shifted to the right, or which amounts to the same thing, upwards. In semiquantitative fashion we can say that collector efficiency η_h and optical efficiency $\eta_{o,h}$ are related

$$\eta_h = F' \, [F_{hot\ mirror} \, \eta_{o,h} - U \, T/I_h] \qquad (II-19)$$

where $F_{not\ mirror}$ is a factor in the range 1.0 to 1.05 which is difficult to measure or calculate. A similar effect exists for flat plate collectors due to the warming-up of the cover glazing(s)[12]. Since $F_{hot\ mirror}$ is difficult to measure as a separate factor it has been absorbed into the definition of $\eta_{o,h}$ throughout the rest of this paper.

Finally the interpretation of test results for concentrating collectors may be obscured by the mixing of optical and thermal effects. In some collectors the receiver may deform at high temperature and move away from its design position. When Eq. II-10 with constant η_o and U is used to fit the resulting data, it falsely ascribes part of the efficiency drop at high temperature to thermal rather than optical losses. Evidence for such an occurance can be seen if the U-value in Eq. II-10 exceeds the heat loss coefficient U_{night} measured at night, or if monitoring of the solar flux near the receiver surface indicates a decreased intercept at high temperature.

III. Testing of the 6.5X

A. Developmental Emphasis

Both collectors underwent extensive performance testing according to the procedures described above. However, since the 6.5X collector was built strictly for experimental purposes a great deal of effort was devoted to its diagnostic testing and improvement modifications. This emphasis is reflected in what follows.

B. Diagnostic Tests

Before constructing the full 7-trough panel a short (∿ 50 cm) length of trough was fabricated and subjected to the preliminary diagnostic tests outlined above in Section II-A. The results are summarized below.

i) Optical Measurements

For the short trough made from aluminized mylar and urethane foam, with a 6.35 mm outer diameter absorber tube, visual inspection yielded a useful* acceptance angle of $2\theta_a = 11°$. This is in acceptable agreement with the design acceptance angle of $2\theta_{a\ \mathrm{design}} = 13°$.

The absorptance of the black chrome absorber[17] tube was measured by Dr. K. Reed of Argonne National Laboratory with a Beckman spectrophotometer and found to be

$$\alpha = 0.96 \qquad \text{(III-1)}$$

For the effective transmittance of the CPC reflector

$$\tau_{CPC} = \rho^{<n>} \qquad \text{(III-2)}$$

the "lightbox" described in Section II A i) yielded a value

$$\tau_{CPC} = 0.85 \qquad \text{(III-3)}$$

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*Angular range over which at least ninety percent of the aperture appeared black.

For a 6.5X CPC with fin or tubular absorber the average number of reflections is

$$\langle n \rangle \approx 1.5 \qquad \text{(III-4)}$$

as can be seen by interpolation from Fig. 11 of Ref. 6. Hence Eq. III-3 is consistant with a reflectance $\rho = 0.90$ for aluminized mylar.

Combining the measured values III-1 and III-3 with the transmittance of the 3 mm thick acrylic cover sheet to be used in the complete collector

$$\tau_{cover} = 0.90 \qquad \text{(III-5)}$$

we predicted an optical efficiency

$$\begin{aligned} \eta_{o,C} &= \tau_{cover}\, \rho^{\langle n \rangle}\, \alpha \\ &= 0.90 \times 0.85 \times 0.96 \\ &= 0.73 \qquad \text{(III-6)} \end{aligned}$$

with respect to radiation within the acceptance angle. Taking $\gamma = 0.90$ to be a typical value for the fraction of clear day hemispherical insolation which is within the acceptance angle of the 6.5X CPC, we thus find that the optical efficiency for the full collector with respect to a pyranometer is expected to range from

$$\eta_{o,h} = \gamma\, \eta_{o,C} = 0.66$$

$$\text{(clear day, i.e. } I_d / I_h = 0.11 \text{ and } \gamma = 0.9) \qquad \text{(III-7)}$$

to

$$\eta_{o,h} = 0.59$$

$$\text{(hazy day, i.e. } I_d / I_h = 0.23 \text{ and } \gamma = 0.80)$$

ii) Heat Loss

For the preliminary tests a short trough module (aluminized mylar on fiberglass-epoxy substrate) was placed in a box 75 cm long, 45 cm wide and 30 cm deep. The space between the module and the walls of the box was filled with insulation and the aperture was covered with an acrylic sheet 3 mm thick in order to simulate conditions in a real collector. This seemingly extravagant amount of insulation at the sides of the CPC module was needed to compensate for the fact that the module was not surrounded by other CPC troughs at the same temperature as would be the case in the complete collector. The absorber tube was heated electrically, and the dissipated power was monitored as well as the temperature at various points of the absorber surface, of the reflector and of the insulation.

A comprehensive theoretical analysis of the expected heat losses was carried out and a detailed set of experiments to determine the relative importance of the various components was performed. Since these methods may be applicable to the general development of nonevacuated CPC's a summary of this study is given in the Appendix. For the net effective heat loss coefficient based on these diagnostic tests we measured

$$U_{\text{total, black chrome, fiberglass}} = 1.88\ \text{W/m}^2{}^\circ\text{C} \qquad \text{(III-8)}$$

using a real absorber tube, coated with black chrome,and a reflecting trough backed with urethane foam insulation. We had no means of measuring the emittance of the black chrome, but from the data we have seen reported for black chrome supplied by Olympic Plating[17], we feel confident that the emittance is in the range of 0.05 to 0.2. Calculation gives a radiative contribution of

$$U_{\text{radiative}, \varepsilon = 0.1} = 0.14 \text{ W/m}^2\text{°C} \tag{III-9}$$

for an emittance of 0.1; hence the radiative contribution to the heat loss coefficient is quite small for the operating temperatures of interest.

Several different techniques were tried for making the CPC reflector. Fiberglass plus epoxy, with either aluminized mylar or vacuum deposited aluminum, turned out not to be practical for a laboratory prototype; the aluminized mylar tended to wrinkle or delaminate from the substrate, and without aluminized mylar the mold release agent prevented a sufficiently smooth surface finish for high reflectivity. As an alternative we poured high temperature urethane foam over aluminized mylar which was stretched over the CPC mold; the results were acceptable and this technique was adopted for fabricating the full panel. The conductive heat losses for a urethane foam reflector are expected to be smaller than for a fiberglass reflector. This was confirmed by a new heat loss measurement, with the same absorber tube as used in the previous test, which yielded

$$U_{\text{total,final version}} = 1.73 \text{ W/m}^2\text{°C} \tag{III-10}$$

iii) Stagnation Tests

These tests were carried out with two perforated metal sheets which transmitted 26% and 51%, respectively, of the incident solar radiation. The ambient temperature was around 20°C and the hemispherical insolation at normal incidence was in the range of 900 to 1000 W/m^2. The insulated module, described in the preceeding subsection was used, with aluminized mylar reflector and urethane foam insulation. The absorber tube temperature was monitored by means of thermocouples inside the tube. The first tests were done with the correct absorber size, i.e., 6.36 mm outer diameter. Since oversized black chrome coated absorber tubes with 9.54 mm outer diameter were also available, we repeated the tests with the latter to evaluate the effect of absorber size on performance. The results are listed in Table I.

Table I. Results of masked stagnation tests for 6.5X.

absorber outer diameter [mm]	$\tau_m I_h$ [W/m^2]	$T_{stagnation}$ [°C]	$T_{ambient}$ [°C]	$U/\eta_{o,h}$ [W/m^2°C]
6.36	241	122	30	2.61
	459	186	33	3.01
9.54	244	112	21	2.67
	472	167	20.5	3.22

These results agree very well with previous diagnostic tests of the 6.36 mm absorber U = 1.73 W/m^2°C of Eq. III-10 and $\eta_{o,h}$ = 0.66 of Eq. III-7 which imply

$$U/\eta_{o,h} = 2.62 \text{ W/m}^2\text{°C}. \qquad \text{(III-11)}$$

Somewhat surprising is the small increase in $U/\eta_{o,h}$ with absorber size, because one would have expected $\eta_{o,h}$ to remain constant and U to increase with absorber size. At high temperature the absorber tube may deform somewhat and miss some of the incident solar radiation in the case of the smaller absorber. Thus the oversized absorber tube can have a significantly higher optical efficiency at high temperature and the ratio U/η_o need not be much worse than for the correct absorber size. Another explanation may be that the dependence of heat losses on a concentration in this configuration and range of concentration ratios is weaker than expected.

C. Performance

The preliminary diagnostic tests revealed no fundamental problems and indeed the measured optical efficiency and heat loss coefficient, consistent with stagnation measurements,were very encouraging. Thus we proceeded with the construction and test of the full 7-trough panel illustrated in Fig. 3a using the procedures described in Section II B. The results are summarized in Figures 5 to 8.

First we determined that the time constant of the collector was very short (less than a minute), as expected from its small heat capacity. Figure 5 shows the collector response when, under full sunshine, a cover is placed over the aperture. The transit time of a fluid element through the the collector was also very short, on the order of two minutes, thus permitting rapid accummulation of data points which is particularly helpful for angular scans.

Figure 6 shows the angular response, that is, the variation of optical efficiency η_o with the incidence angle $\theta_\perp$ projected transverse to the trough. These data points were obtained by pointing the collector normal to the sun and then changing its tilt up and down. The dashed line shows the behavior expected if the optics were perfect. The efficiency is nearly constant throughout the central portion of the design acceptance angle $2\theta = 13°$, dropping off near the edge, i.e., between 5 and 7 degrees from the center.

To evaluate long term average energy delivery one also needs to know the variation of $\eta_{o,h}$ with $\theta_{||}$, the projection of the incidence angle along the

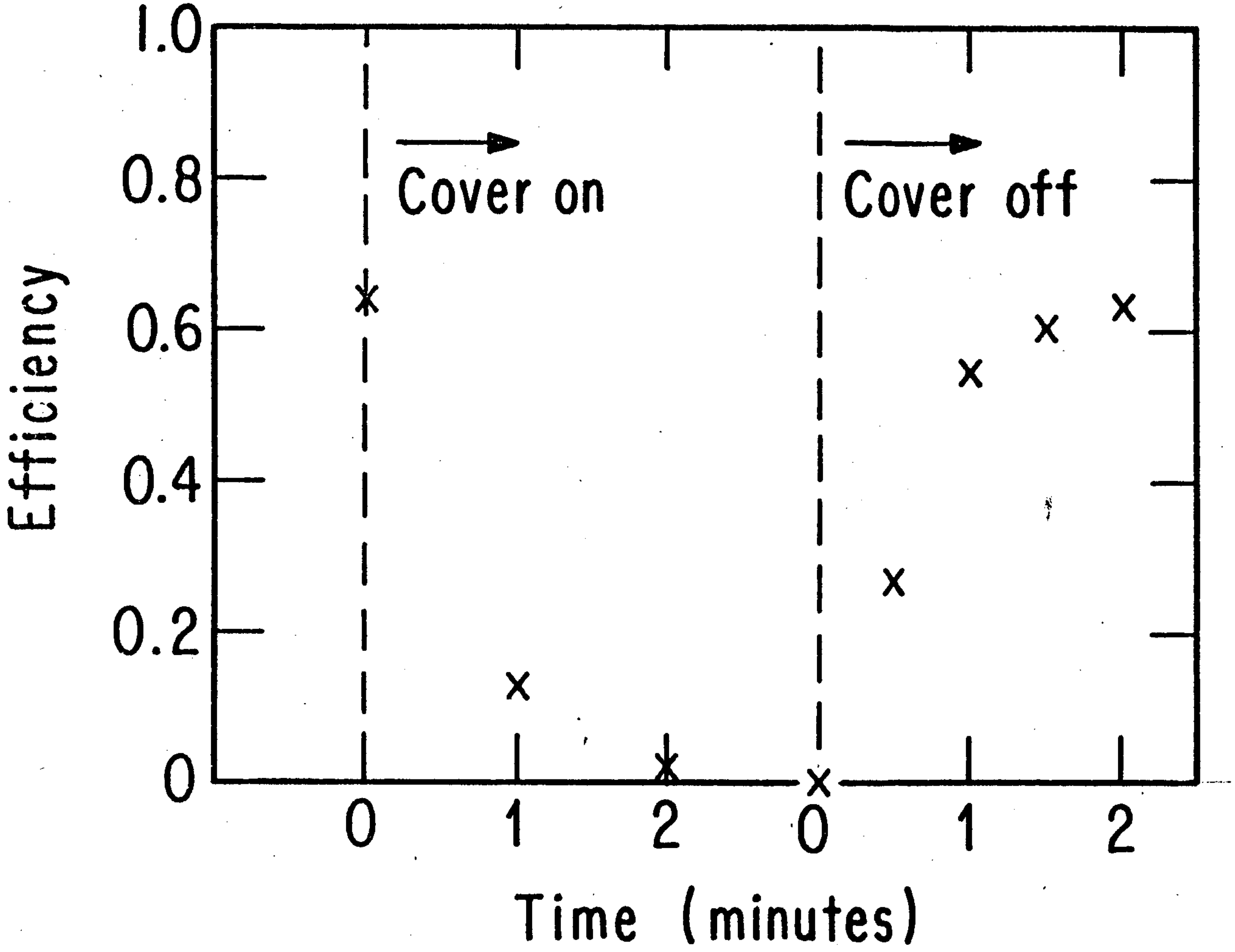

Fig. 5 Transient Response of 6.5X CPC.

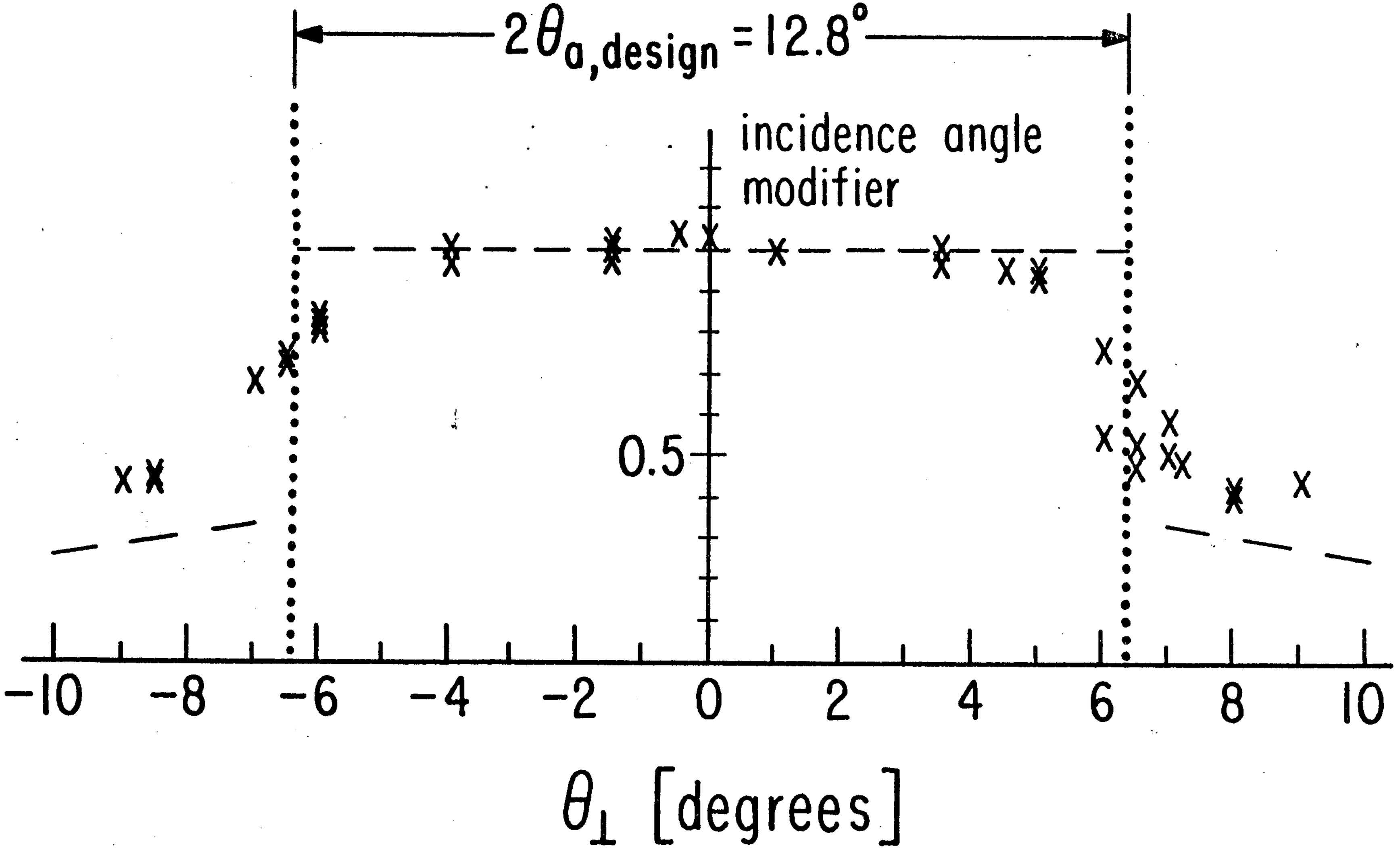

Fig. 6 Incidence angle modifier $\kappa(\theta_\perp, \theta_{||} = 0)$ for 6.5X CPC.

trough (east-west). For this purpose the collector was placed in a fixed position (due south, and tilted so that the sun at noon is just within the acceptance angle), and it was operated for the whole day. This test was run near solstice and hence $\theta_{||}$ was nearly equal to the hour angle; the variation of $\theta_{\perp}$ with $\theta_{||}$ was negligible at this time of year. The measured optical efficiency is plotted versus $\theta_{||}$ in Fig. 7. The apparent morning-afternoon asymmetry was caused by "closed loop" operation. Even though the time constant of the collector itself is short,the input of solar energy caused a slow rise of the temperature of the fluid in the test loop and thus prevented the attainment of steady state conditions during most of the day. (That is why we now recommend open loop operation for this test.) By averaging morning and afternoon data, one can partially compensate for this transient effect.

The measured instantaneous thermal performance is shown in Fig. 8. The data points correspond to individual efficiency measurements plotted as a function of $\Delta T/I_h$. The solid line is a least squares fit to the data assuming the form given by Eq. II-10 and corresponds to values for the collector parameters of

$$F'\eta_{o,h} = 0.65 \pm 0.02 \qquad \text{(III-12)}$$

and

$$F'U = 2.2 \pm 0.2 \text{ W/m}^2\text{°C} \qquad \text{(III=13)}$$

Also shown in the figure is a dotted line corresponding to the expected performance based on the preliminary diagnostic tests.

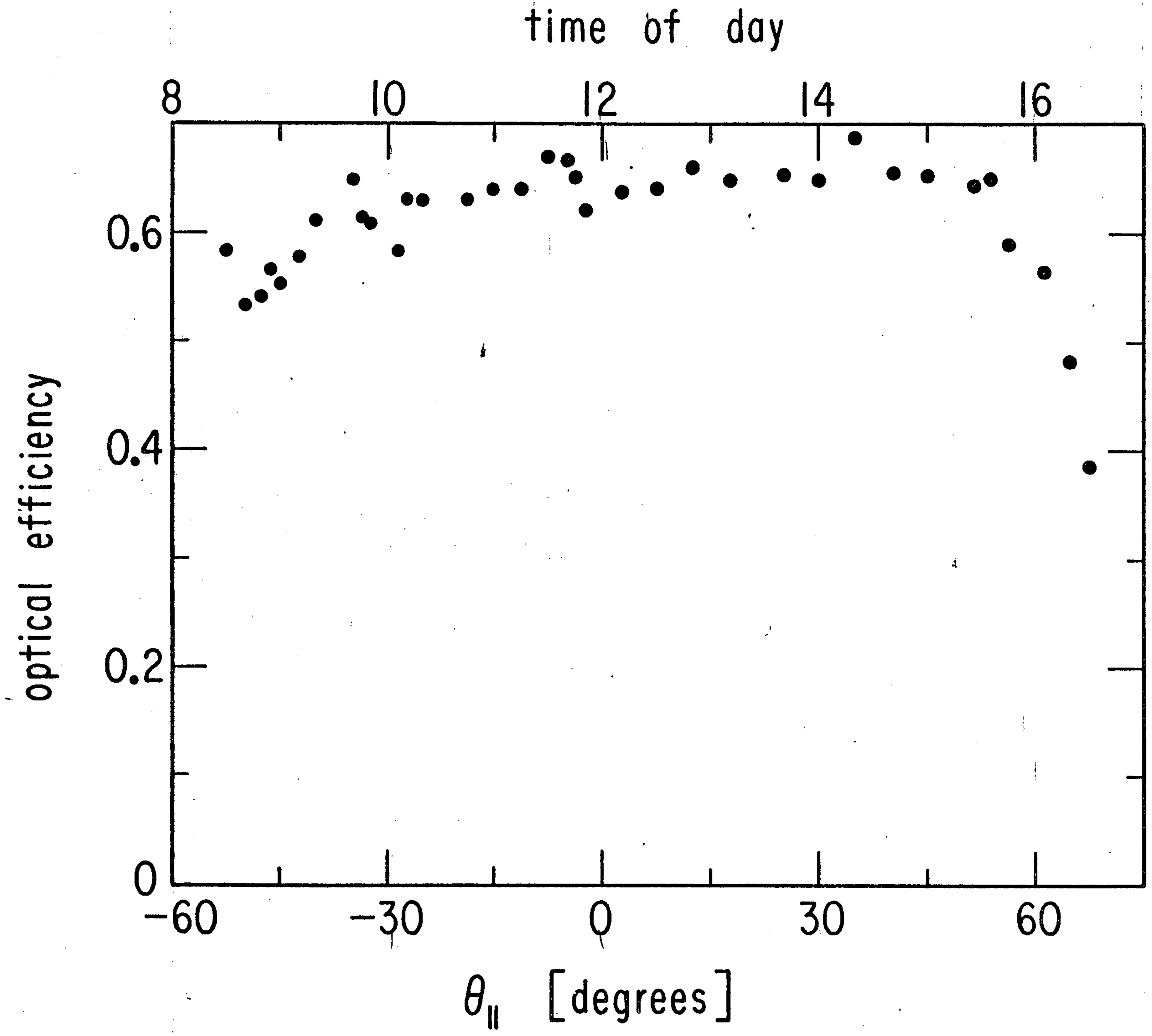

Fig. 7 Optical efficiency versus incidence angle $\theta_{||}$ along trough.

The uncertainties in the measured collector parameters above are based on our best estimate of the errors arising from a) variations in the relative fraction of diffuse insolation, b) uncertainites in the absolute value of I_h due to non-horizontal orientation of the Eppley pyranometer used and c) possible slight deviations from stable thermal equilibrium of the collector during the measurements. That is, the standarized correction for diffuse insolation and the use of "open loop" flow tests were not part of our procedure at that time. These factors may account for some of the discrepancies between the expected performance (dashed line) and observed (solid line). However, even taking these uncertainties into account and allowing for some additional heat loss due to the interconnecting tubes between the troughs, it can be seen that the optical efficiency is slightly lower and the effective thermal loss somewhat higher than expected based on diagnostic measurements on the prototype trough.

One possible explanation for these effects has to due with the "gap losses" associated with misalignment of the absorber tubes. These tubes in the 6.5X version were made of copper and tended to be slightly bent even after the initial installation. It has been calculated[8] that a 1 mm lateral displacement of a 6 mm diameter tube can cause a 10% optical loss. This effect can be enhanced at high temperatures if the thermal expansion of the tubes causes them to become further warped. This would result in a thermally dependent optical loss so that the thermal performance of the collector would be characterized by a spuriously high U-value. There is some evidence that some of the poorer than expected performance of the 6.5X was due to just such an effect since visual inspection of the collector aperture showed that it looked less dark after the high temperature testing had been completed.

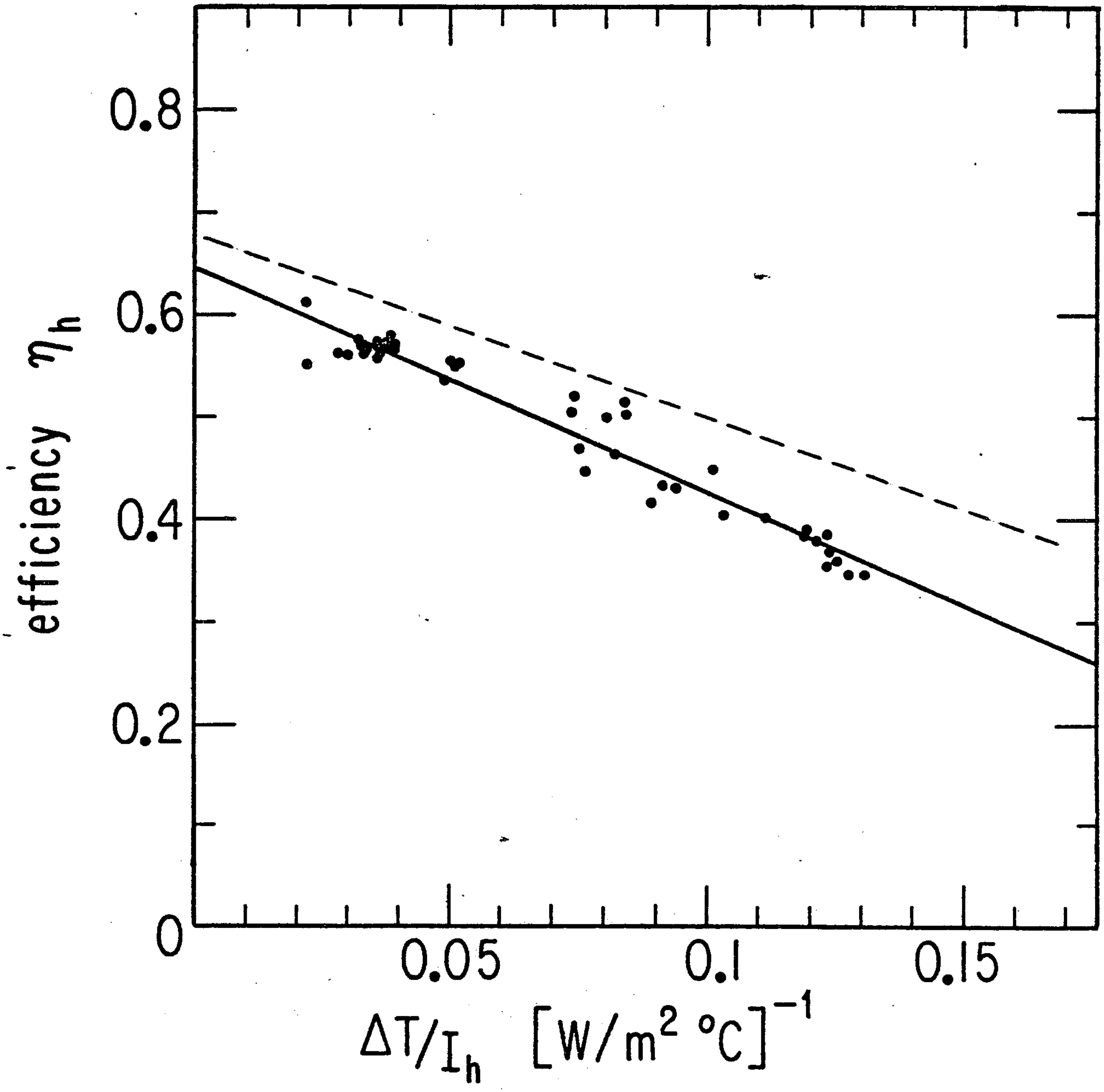

Fig. 8 Instantaneous efficiency of 6.5X CPC: fit to data (solid line), expected performance after preliminary tests (dotted line).

To avoid such sensitivity to misalignment and thermal deformation, particularly in the cases where high concentration ($\gtrsim$ 4X) is combined with absorbers of small absolute dimensions ($\lesssim$ 1.25 cm), we strongly recommend a) the use of absorbers oversized by about 20% with respect to the "theoretical" mathematical absorber surface for which the reflector profile surface is generated and b) the use of steel or other relatively strong material for the absorber tubes.

Following this prescription we decided, as final phase of this development effort, to refurbish the 6.5X collector with larger steel absorber tubes of 7.9 mm outer diameter (5/16 in) instead of the original 6.4 mm (1/4 in) copper tubes and measure its thermal performance as described in the next section.

D. Revised Version, the 5.2X

Since the reflector profile and aperture are unchanged and only the absorber tube diameter is increased from 6.4 to 7.9 mm the concentration ratio is reduced from 6.5 to 5.2 and it is more appropriate to refer to the modified collector as a 5.2X.

The measured efficiency is plotted in Fig. 9. This time scatter of the data points due to variable atmospheric haze has been reduced by correcting the measured efficiency and insolation values according to Eq. (II-8) with a reference ratio $I_d/I_{h,\text{standard}}$ = 0.11, the low temperature points were measured in an "open loop" flow configuration and the high temperature points were obtained after carefully attaining a condition of temperature equilibrium for the full collector. The solid line is a least squares straight line fit to the data and corresponds to the collector parameters

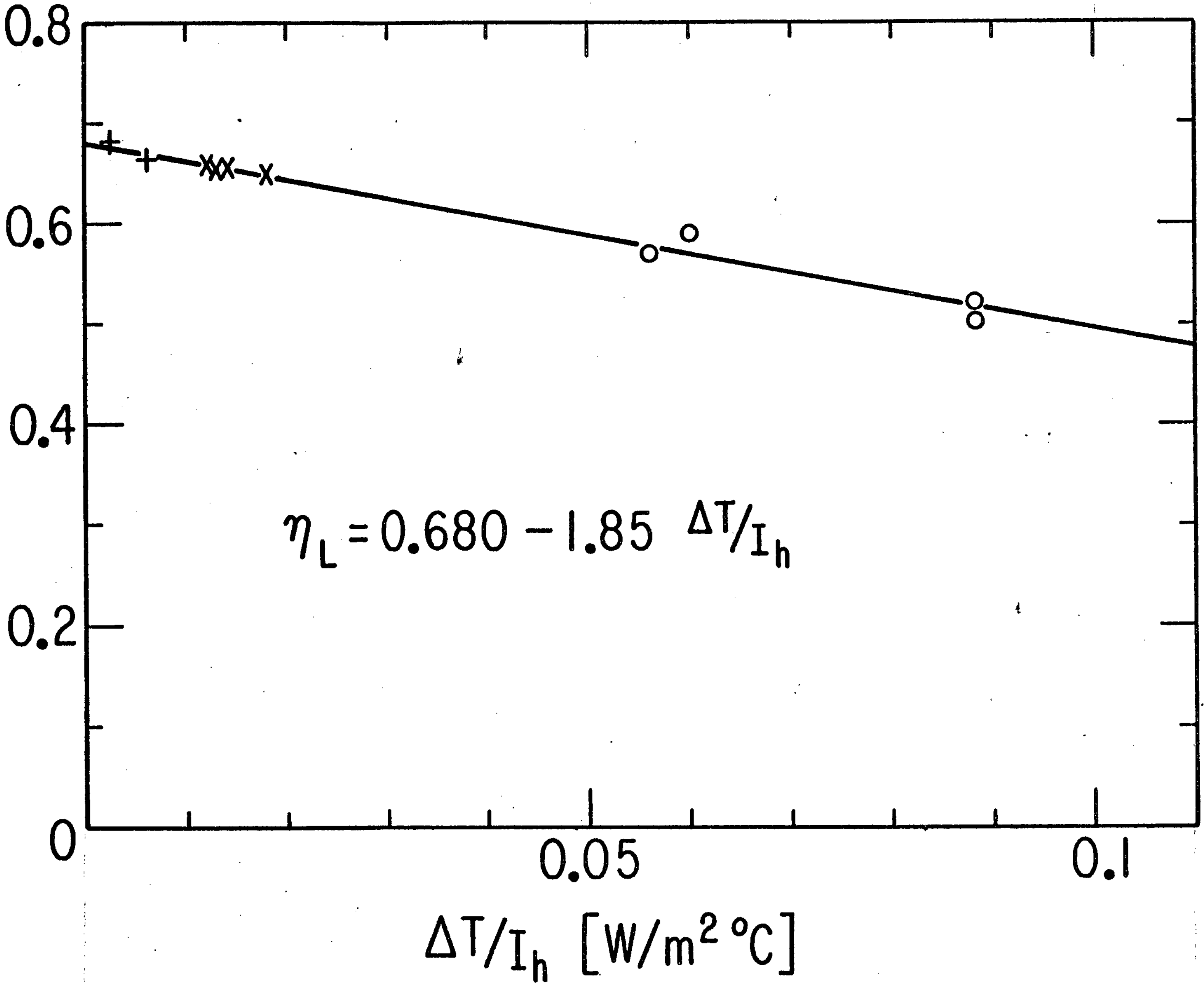

Fig. 9 Instantaneous efficiency of 5.2X CPC.

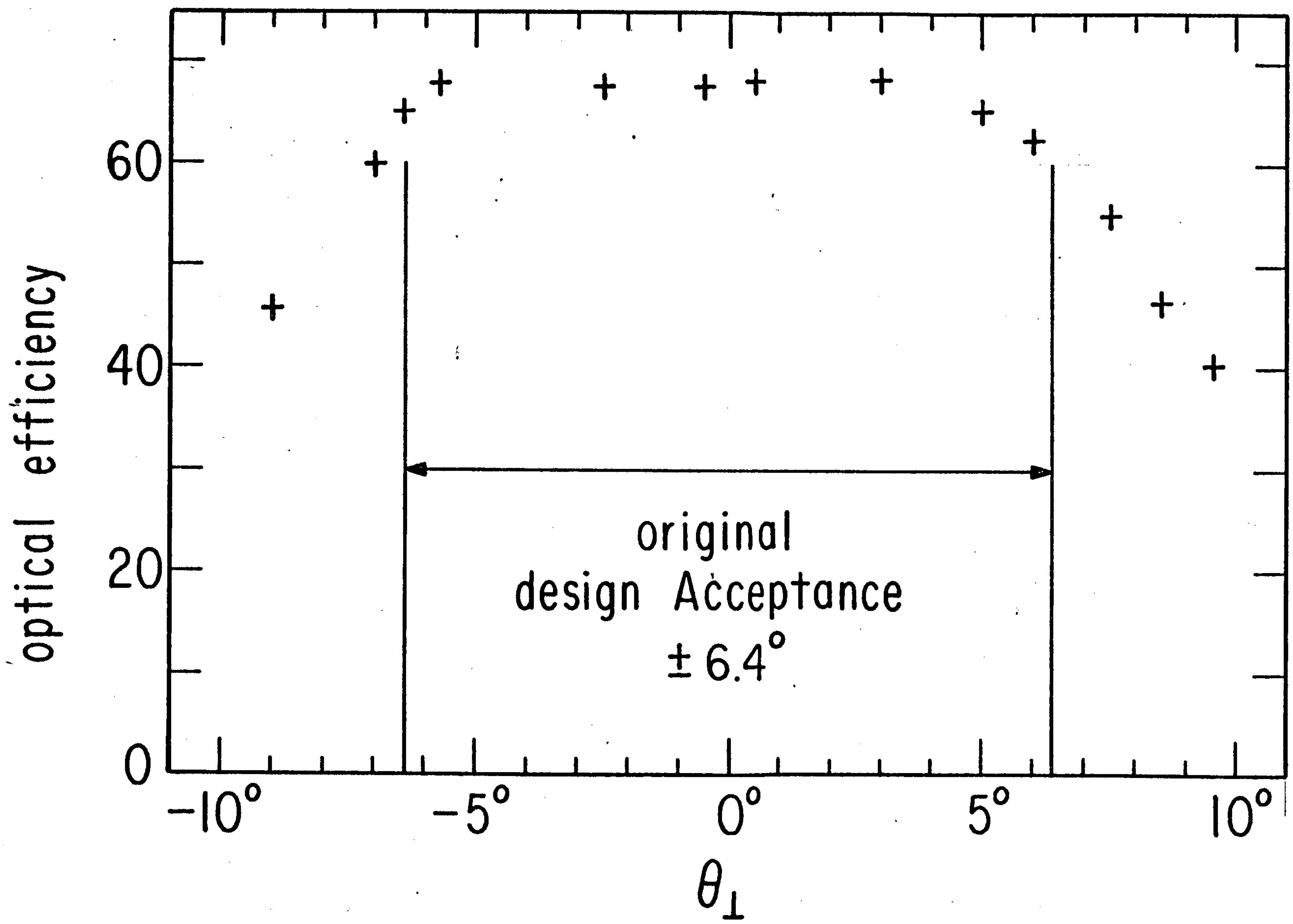

Fig. 10 Optical efficiency of 5.2X versus $\theta_\perp$ (transverse to trough at $\theta_{||} = 0$).

$$F'\eta_h(0) = 0.68 \pm 0.01 \qquad \text{(III-23)}$$

and

$$F'U = 1.85\ \mathrm{W/m^2\,°C} \pm 0.1 \qquad \text{(III-24)}$$

These are the data points which have been shown in Fig. 1, superimposed on the performance prediction of several years ago[6]. They agree very well with the prediction implied by the C = 4 and C = 8 curves for CPC's with selective absorber. This close agreement is fortuitous in the sense that the material properties which had been assumed do not all agree exactly with those of the actual collector; they are, however, quite close and do represent the collector on the average.

The angular scan in the transverse plane for $\theta_\perp$ for the 5.2X is shown in Fig. 10. The region where $\eta_h(0)$ can be assumed constant has been enlarged slightly, by about half a degree compared to the original 6.5X (exact comparison is difficult for lack of data in the region around $\theta_\perp = 6.4°$). Most of the gain in angular acceptance associated with reduced concentration has occured outside the design acceptance angle and is not very useful.

The improved thermal performance of the 5.2X modification shown in Fig. 9 compared to the original 6.5X in Fig. 8 represents both the improvements associated with the oversized tubes and the refinement of our test procedure, in particular the standardization of the efficiency measurement to a very clear day value of $I_d/I_h = 0.11$. The final achievement of good agreement with the predicted performance based on a theoretical model (Fig. 1) and with the experimental measurements from the preliminary diagnostic tests shows clearly that with sufficient care one can design and build a non-evacuated CPC with excellent performance at intermediate temperatures.

IV. Testing of the 3X

A. Production Emphasis

In contrast to the 6.5X which was a fully experimental collector development, the 3X collector was conceived, built and tested as a field collector for use in a relatively large array (73 m^2). Therefore the diagnostic testing and development procedures were not carried out in the same detail.

B. Preliminary Diagnostic Testing.

For this purpose a single trough prototype module consisting only of the absorber fin and CPC reflector (profile configuration shown in Fig. 1b) was built. The reflector was fastened to wooden ribs and mounted within a plywood box with an acrylic cover glazing.

i) Optical characteristics: the visual inspection method was applied only to evaluate the quality of the mirror contour by making sure there were no reflecting shiny patches when the aperture was viewed from a distance. A quantitative measurement of the acceptance angle was not carried out. The optical parameters of the components were taken to be a transmittance of

$$\tau_{cover} = 0.90$$

for the acrylic glazing and a reflectance of

$$\rho = 0.84$$

for the sheet aluminum reflectors (Kinglux$^{(R)}$) with an average number of reflections

$$<n> = 1.25 .$$

With a black chrome plated absorber fin with absorptance

$$\alpha = 0.94$$

and a correction factor $\gamma = 0.93$ for loss of diffuse, standardized to a diffuse fraction of 0.11, one expects an optical efficiency of

$$\eta_{o,h} = \tau_{cover} \cdot \rho^{<n>} \cdot \alpha \cdot \gamma$$

$$= 0.63$$

Note that if the actual diffuse fraction under hazy test conditions is as high as 0.23 the loss of diffuse correction is 0.85 so the expected optical efficiency may be as low as

$$\eta_{o,h} = 0.58 \text{ (hazy day).}$$

ii) <u>Heat loss tests</u>; the heat loss coefficient of the prototype trough in the wooden box with essentially only dead air space in back of the reflector as an insulation was measured using an electrical resistance heat source along the length of the inside of the absorber tube. A value of

$$U_{lab} = 3.0 \text{ W/m}^2\text{°C}$$

was found. It is interesting to note that later tests with the full prototype unit in which the reflector was backed by urethane foam insulation yielded values only 10% lower than this and as a result the full production version was in fact fabricated using only dead air space as insulation.

iii) <u>Stagnation tests</u>; both the wooden prototype unit and both troughs of the full two-trough prototype collector were subjected individually to stagnation tests under full solar insolation levels. Values of the stagnation temperature ranging from 200°C to 220°C above ambient were observed corresponding to values of $U/\eta_o = 4.6 \pm 0.3$ W/m^2°C. This is to be compared with the expected values for this ratio of 4.76 based on the numbers for η_o and U discussed in i) and ii) above.

C. Performance Testing

The optical and thermal performance of the full two trough module was measured using the same procedures as used for the original 6.5X collector and unfortunately did not utilize some of the refinements and standardization procedures which we developed at a later time.

The incidence angle modifier as a function of $\theta_{\perp}$, the incidence angle projected into the transverse plane, is shown in Fig. 11. Note that the output is relatively flat within ±16° of the aperture normal and it drops off to no less than 80% of full response at ±20°. The design acceptance angle was ±18°. It should be noted that the cover glass used for the collector module here was the ASG water-white glass which has a slight surface texturing resulting in a blurring of transmitted images. The effect of the scattering caused by this texturing is only to contribute to the rounding of the response profile in the "shoulders" between ±16° and ±20° and not to degrade the optical performance of the concentrator in any significant way. This can be understood in terms of the fact that the magnitude of scattering of transmitted light rays is of the order of 1° - 3° which is a small fraction of the design acceptance angle.

The incidence angle modifier in the longitudinal plane was not measured for the prototype since this will be measured in the field for elements of the actual array whose long term performance and comprehensive angular response characteristics will be the subject of a forthcoming study.

The thermal performance of the 3X collector is shown in Fig. 12 as a function of $\Delta T/I_h$. The solid line represents a least squares fit to the

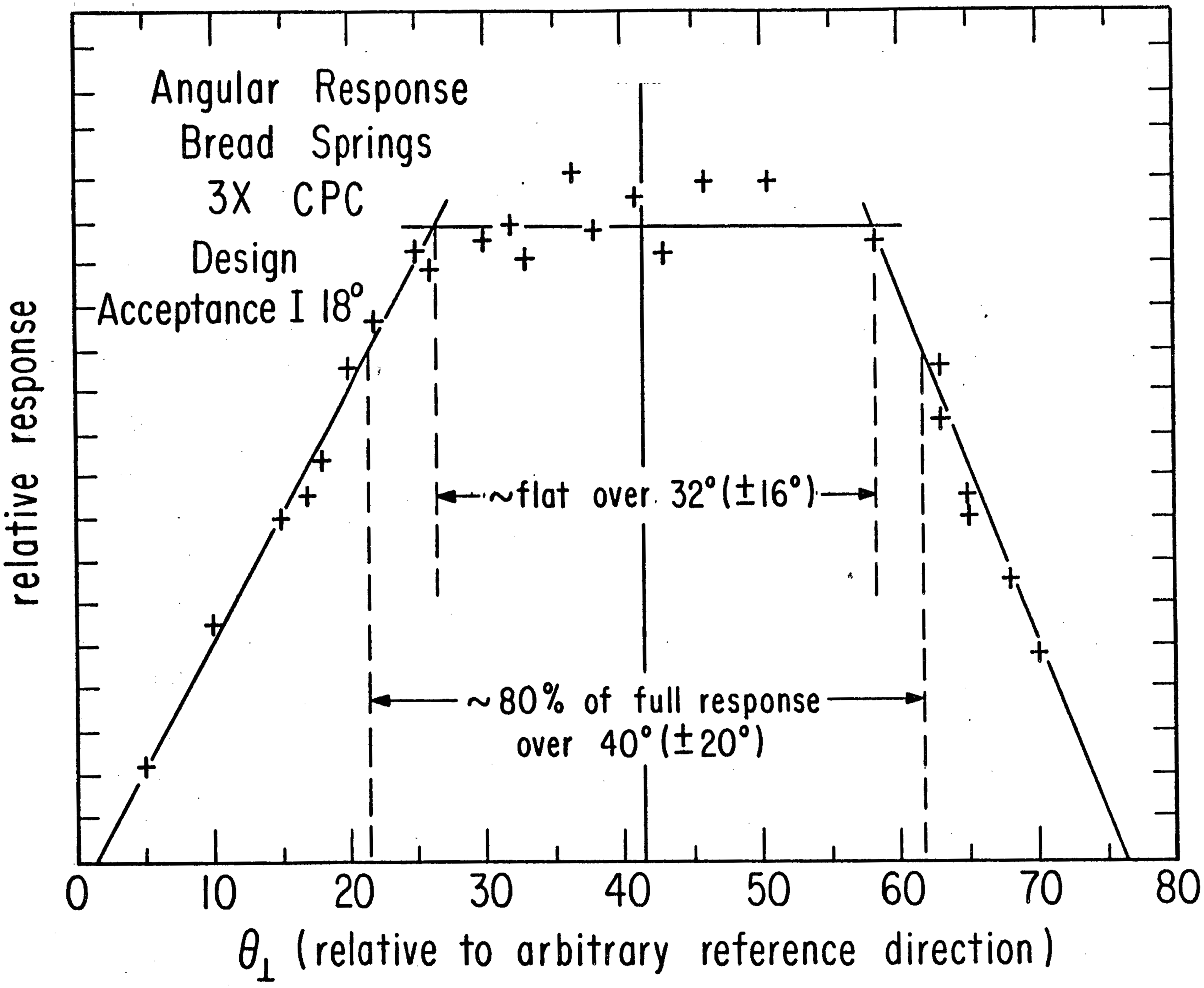

Fig. 11 Optical efficiency of 3X versus $\theta_{\perp}$.

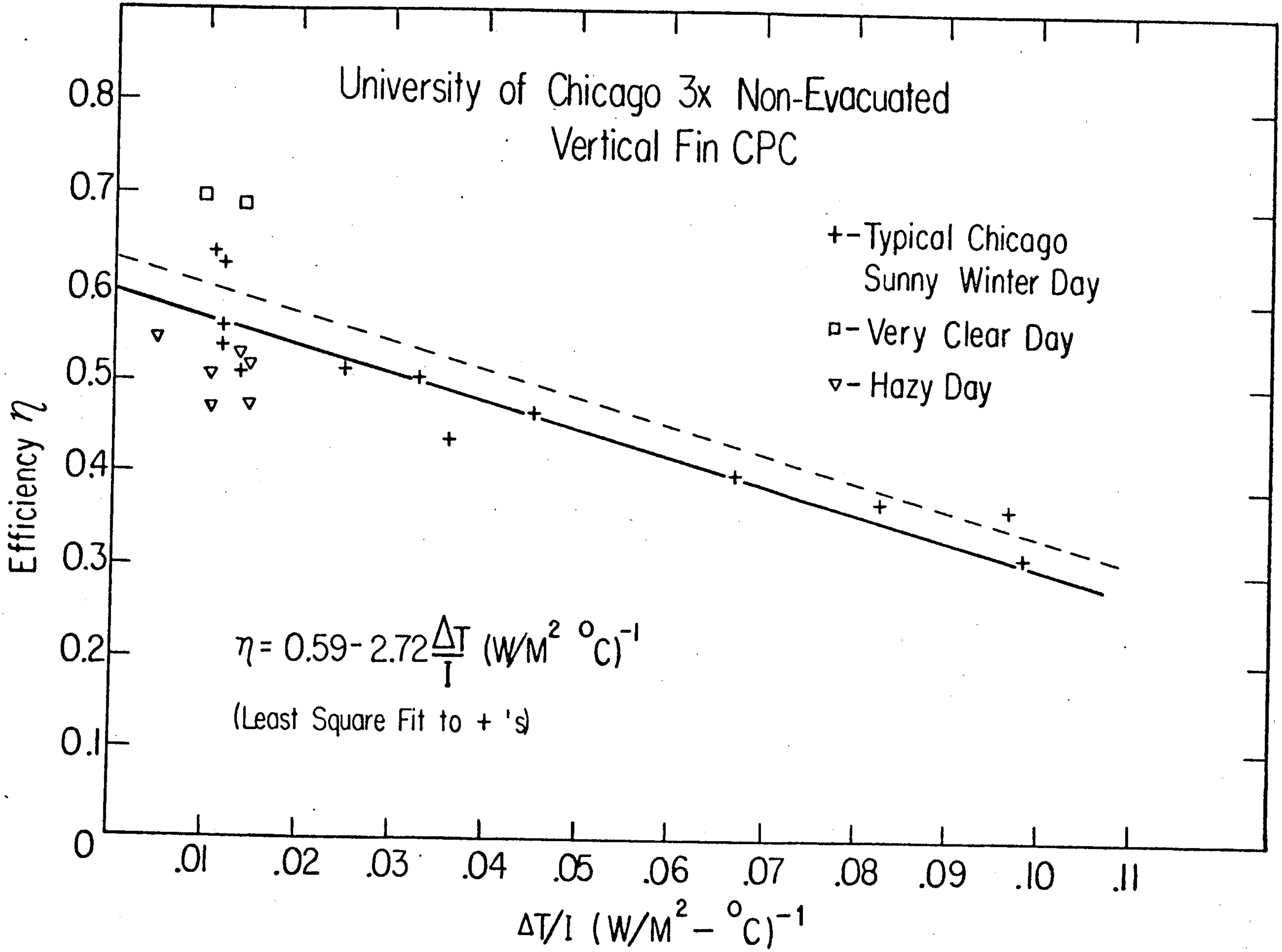

Fig. 12 Instantaneous efficiency of 3X.

efficiencies measured under typical clear day conditions and as in the case of the 6.5X, the ~~dotted~~ dashed line represents the expected performance based on preliminary diagnostic testing. The collector parameters corresponding to the fit in the figure are

$$F'\eta_{o,h} = 0.59 \pm 0.02$$

and

$$F'U = 2.7 \pm 0.2 W/m^2 {}^\circ C$$

The heat loss coefficient is in good agreement with expectation as is the optical efficiency when consideration is made for the probable effect of somewhat more "loss of diffuse" than under the standardized clear day conditions. Although quantitative information on this correction is lacking for the time period of the measurement (spring of 1977) this interpretation is born out by two features: a) the "very clear day" points shown as open squares in Fig. 12 which correspond to an optical efficiency above 65% and b) the "typical Chicago winter day" conditions we now know correspond to a diffuse/total fraction of ∿0.16. This in turn corresponds to a predicted optical efficiency of 0.60 which is precisely in agreement with the observations. All of these observations are consistent with a value of

$$F'\eta_{o,h} = 0.63 \pm 0.02,$$

when referred to the standardized clear sky conditions of a diffuse/total fraction of 0.11.

V. Conclusions

The instantaneous efficiency results for the nonevacuated CPC collectors described in this paper are summarized in Fig. 13 and compared to a flat plate collector. The flat plate,

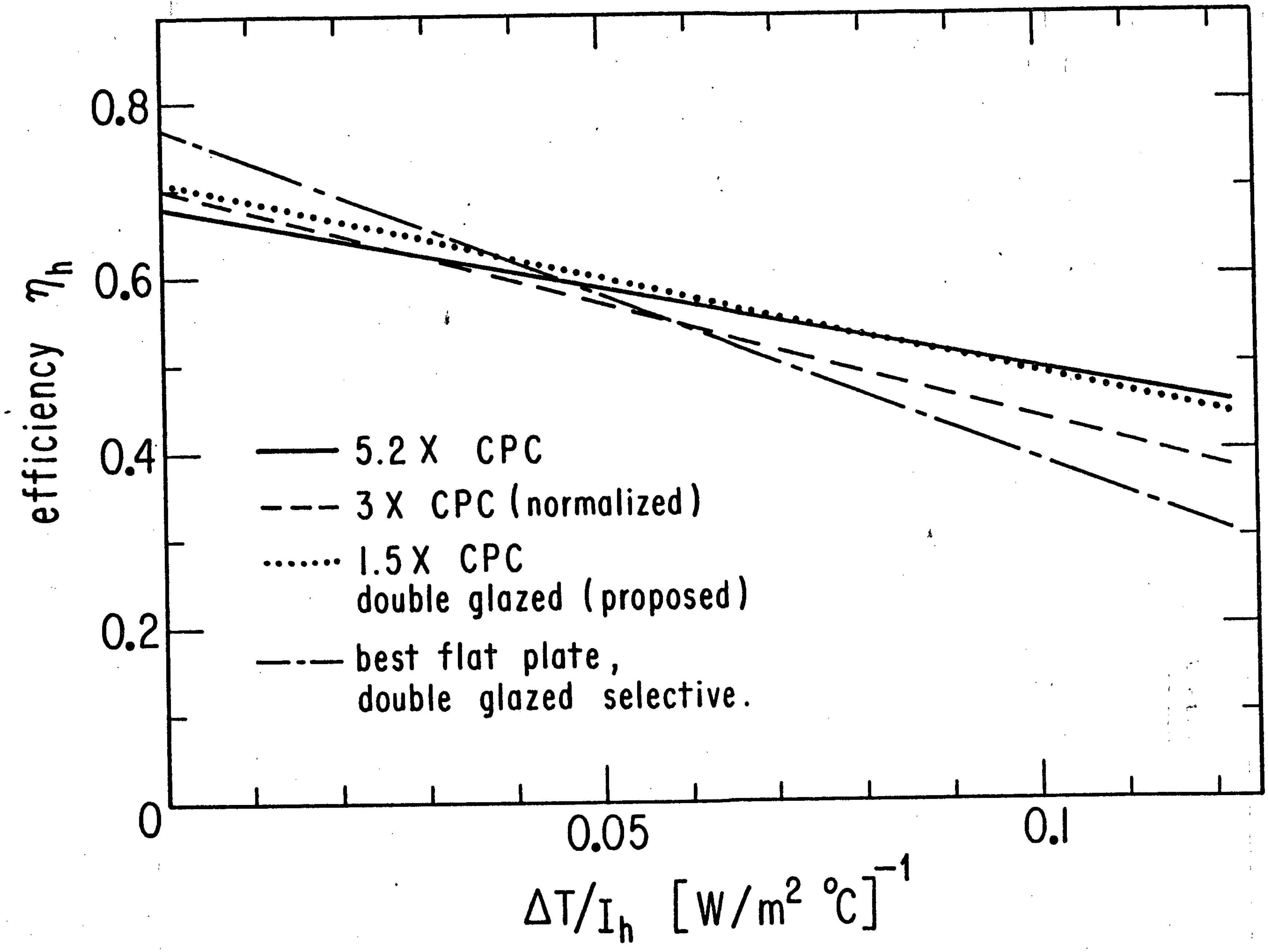

Fig. 13 Comparison of instantaneous efficiency curves.

double glazed and selective, is one of the very best available[20]. On this graph the 5.2X outperforms the best flat plates when the mean fluid temperature is more than approximately 40°C above ambient. The 3X, corrected to standardized conditions and normalized to an optical efficiency corresponding to materials identical to the 5.2X would outperform the very best flat plate above about 60°C above ambient. This is particularly important when it is recognized that this 3X could be fabricated very economically since the absorber is only a 6.4 cm (2.5 in.) fin and no insulation other than dead air between the reflector and outer housing was used in the production version installed at the Bread Springs School.

Consideration of the heat transfer from absorber to ambient shows that glazing in a concentrating collector is more effective as thermal barrier if placed close to the absorber rather than at the aperture. On the other hand, in a CPC trough aligned in the east-west direction a cover at the aperture appears to be necessary in order to protect the reflector from excessive dirt accumulation. Therefore a glass envelope around the receiver of a CPC would be in addition to the cover glazing, and we shall refer to such a design as double glazed CPC. A stationary double glazed CPC of concentration C = 1.5 is sketched in Fig. 14. The extra glass to air interfaces cause optical losses which we thought to be excessive when we designed the 3X and the 6.5X (5.2X). At the time higher concentration seemed to be the better approach. Since then, however, low cost antireflection coatings have become available even for tubular glass, and with that a double glazed 1.5X CPC is preferable to a single glazed 5.2X.

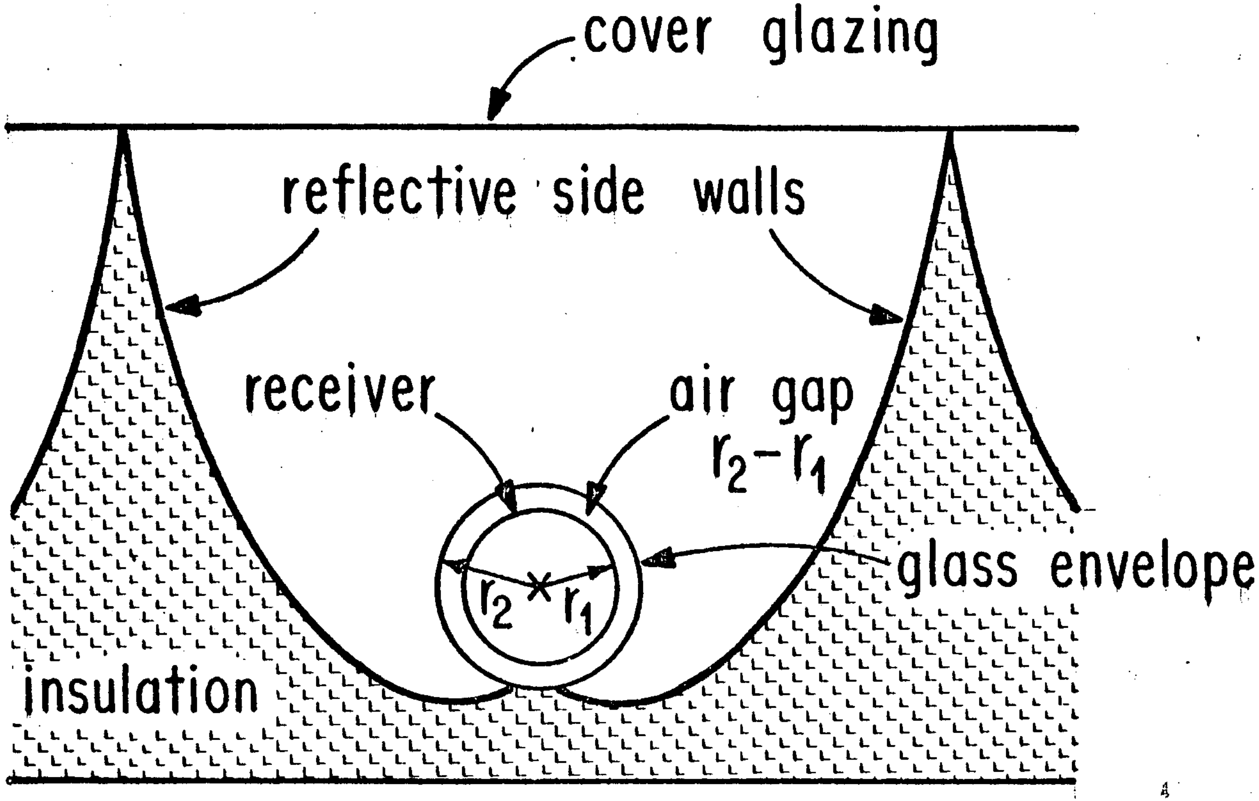

Fig. 14 CPC reflector to tubular non-evacuated receiver.

To analyze the performance expected of an advanced double glazed 1.5X, let us assume the following optical properties

τ_1 = 0.95 = transmittance of cover

$\rho^{<n>}$ = 0.95 = throughput of CPC with second surface silver reflectors

τ_2 = 0.92 = transmittance of receiver envelope

α = 0.94 = absorptance of receiver

γ_{gap} = 0.94 = 1 - losses in gap between envelope and absorber.

The product of these numbers yields the optical efficiency with respect to radiation within the acceptance angle as $\eta_c = 0.734$. For comparison with flat plates this is converted to

$$\eta_h = 0.711$$

according to Eq. II-14. The heat loss can be calculated by the methods of Ref. 6. Choosing the width of the air layer between absorber and inner surface of the glass envelope involves a compromise because for a small gap conduction is large whereas a large gap implies high optical losses. The air space should not be increased beyond the threshold for the onset of natural convection; this corresponds to a Raleigh number of approximately 1000, with the width $r_2 - r_1$ of the gap as characteristic dimension. For the operating temperature under consideration this Raleigh number corresponds to a gap of $r_2 - r_1 \stackrel{\sim}{\sim} 0.8$ cm. This condition can easily be met without excessive optical losses in the gap, because tube radii on the order of 2.5 cm are practical for such a collector, and the resulting optical loss is on the order of 6% only, as assumed above.

When the heat loss is calculated for such a configuration, one obtains a U-value around 2.2 W/m^2K. The performance of such a collector is shown by the dotted line in Fig. 11. It is slightly better than the 5.2X and does not require any tilt adjustments.

Acknowledgements

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Nonmenclature

C = geometric concentration (= ratio of aperture area over absorber surface area)

c_p = heat capacity

F' = collector efficiency factor (see Ref. 12 & 13) which accounts for difference between mean fluid temperature and absorber surface temperature

I_b = the beam component of solar irradiance [W/m^2]

I_C = solar irradiance [W/m^2] within acceptance angle of concentrating collector

I_d = diffuse irradiance

I_h = hemispherical (also called total or global) irradiance

$\dot{m}$ = flow rate [kg/sec]

$\langle n \rangle$ = average number of reflections

q = net heat output of collector [W]

$T_{ambient}$ = ambient temperature

T_f = mean fluid temperature

ΔT = $T_f - T_{ambient}$

U = heat loss coefficient [$W/m^2 {}^\circ C$] relative to collector aperture area

α = solar averaged absorptance

ε = emittance

η_h = efficiency with respect to I_h (pyranometer)

η_C = efficiency with respect to I_C

$\eta_{o,h}$ = optical efficiency with respect to I_h

$\eta_{o,C}$ = optical efficiency with respect to I_C

ρ = solar averaged reflectance

τ = solar averaged transmittance

τ_{CPC} = solar averaged throughput of CPC reflector

$\theta_{||}$ = projection of incidence angle along trough

$\theta_{\perp}$ = projection of incidence angle perpendicular to trough

θ_a = acceptance half angle

$\kappa(\theta_{||},\theta_{\perp}) = \eta_o(\theta_{||},\theta_{\perp})/\eta_o(\theta_{||}= 0,\ \theta_{\perp}= 0)$ = incidence angle modifier

Appendix

Heat Loss Analysis

This series of measurements was concerned with comparison between data and analytical predictions for the heat loss in the CPC configuration. Theoretical treatment of the problem is difficult because correlations for natural convection in CPC enclosures are not known[19]; furthermore, there is strong coupling between conductive, convective and radiative heat transfer modes. When the absorber is selective, natural convection, which is most difficult to calculate, is expected to dominate. In view of this situation we have shied away from an attempt to improve upon the simple first order estimate discussed in Ref. 6. In that paper conduction, convection and radiation were assumed to be decoupled from each other, and convection was calculated by means of the well known correlations for natural convection from flat surfaces[14]. Whether the length or the width of the absorber is to be chosen as characteristic dimension, depends on the nature of the convection currents (longitudinal or transverse). For turbulent convection this does not matter because the heat transfer coefficient is scale invariant. For laminar convection on the other hand, the heat transfer coefficient varies like the 1/4 power of the characteristic length. We have therefore calculated the convective heat transfer for both cases; the difference, about forty percent, gives an indication of the reliability of this naive model. We also injected smoke into the CPC to observe the flow patterns: with horizontal CPC aperture we found longitudinal convection currents, but with the aperture at typical tilt angles (30° to 60°) the flow took on a mixed or spiraling pattern.

To learn more about the magnitude of the convection heat transfer we replaced the absorber of the test module described in Section III B ii) by a well polished aluminum tube to minimize radiative effects. For this measurement the first prototype reflector, made of fiberglass and epoxy with vacuum deposited aluminum, was used and the insulation between reflector and box was fiberglass (conductivity estimated at k = 0.005 W/m °C). A total U-value of

$$U_{\text{aluminum, total}} = 1.86 \ \text{W/m}^2\,^\circ\text{C} \tag{A-1}$$

was measured at

$$\Delta T = T_{abs} - T_{amb} = 93^\circ\text{C}$$

The accuracy of the heat loss measurements is about 0.02 $\text{W/m}^2\,^\circ\text{C}$. Then the reflector was filled with fiberglass and a new U-value of

$$U_{\text{aluminum, filled}} = 1.11 \ \text{W/m}^2\,^\circ\text{C} \tag{A-2}$$

was found at the same ΔT. From the geometry we estimate that about seven-eighths of this is due to conduction through back and sides

$$\tilde{U}_{\text{aluminum, back}} = 0.88 \ U_{\text{aluminum, filled}} \tag{A-3}$$

The tilde indicates that this is not the real back loss coefficient because the reflector temperature was different between these two measurements even though the receiver temperature was the same. With insulation inside the reflector, the CPC wall is not cooled by convection currents and reaches higher temperatures. We have therefore rescaled $\tilde{U}$ by the factor

$$\frac{T_{\text{reflector}} - T_{\text{ambient}}}{T_{\text{reflector,filled}} - T_{\text{ambient}}} = 0.79 \qquad \text{(A-4)}$$

taking the measured temperature at the bottom of the reflector as relevant reflector temperature. Combining the correction factors (A-3) and (A-4) we obtain the backloss coefficient

$$\begin{aligned} U_{\text{back}} &= 0.88 \times 0.79 \times U_{\text{aluminum,filled}} \qquad \text{(A-5)} \\ &= 0.77 \text{ W/m}^2{}^\circ\text{C} \end{aligned}$$

with an estimated error of ±0.1 $\text{W/m}^2{}^\circ\text{C}$.

Hence the front loss is

$$\begin{aligned} U_{\text{aluminum,front}} &= U_{\text{aluminum,total}} - U_{\text{back}} \\ &= 1.86 - 0.77 \text{ W/m}^2{}^\circ\text{C} \\ &= 1.09 \text{ W/m}^2{}^\circ\text{C} \qquad \text{(A-6)} \end{aligned}$$

Our lack of knowledge about the precise value of the emittance of the aluminum surface does not matter because the radiative contribution to $U_{\text{aluminum,front}}$ is calculated to be small, about 0.05 $\text{W/m}^2{}^\circ\text{C}$ for an estimated emittance of 0.04. The radiative contribution is comparable to the uncertainty of the measurements. Therefore we conclude that the convective contribution to the U-value of the 6.5X CPC is

$$\begin{aligned} U_{\text{convective}} &= U_{\text{aluminum,front}} - U_{\text{aluminum,radiation}} \\ &= 1.09 - 0.05 \\ &= 1.04 \pm 0.1 \text{ W/m}^2{}^\circ\text{C} \qquad \text{(A-7)} \end{aligned}$$

By comparison, the calculation based on the naive model discussed in Ref. 6 and outlined above yields

$$U_{\text{convective,theory}} = 1.07\ \text{W/m}^2{}^\circ\text{C for liminar convection} \quad \text{(A-8)}$$

and

$$U_{\text{convective,theory}} = 0.70\ \text{W/m}^2{}^\circ\text{C for turbulent convection} \quad \text{(A-9)}$$

We note that the measured convective heat loss agrees with the higher of the two estimates provided by the theoretical model of Ref. 6.

Since the back loss coefficient of 0. 77 W/m^2°C was larger than desired, we replaced the fiberglass insulation in the test box by polyurethane foam. Repeating the measurements and data correction procedure above we found that the back loss coefficient had decreased to

$$U_{\text{back,foam}} = 0.51\ \text{W/m}^2{}^\circ\text{C}, \quad \text{(A-10)}$$

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19. Recently an exact theoretical analysis of natural convection in CPC enclosures has been published, but it considers only the CPC configuration with flat one-sided absorber, not the collector configurations of the present paper.

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