EXPERIMENTAL EVALUATION OF SOLAR THERMOSYPHONS WITH HEAT EXCHANGERS

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by

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SOLAR BUILDINGS RESEARCH AND DEVELOPMENT PROGRAM
CONTEXT STATEMENT
November 21, 1985

In keeping with the national energy policy goal of fostering an adequate supply of energy at a reasonable cost, the United States Department of Energy (DOE) supports a variety of programs to promote a balanced and mixed energy resource system. The mission of the DOE Solar Buildings Research and Development Program is to support this goal, by providing for the development of solar technology alternatives for the buildings sector. It is the goal of the program to establish a proven technology base to allow industry to develop solar products and designs for buildings which are economically competitive and can contribute significantly to building energy supplies nationally. Toward this end, the program sponsors research activities related to increasing the efficiency, reducing the cost, and improving the long-term durability of passive and active solar systems for building water and space heating, cooling, and daylighting applications. These activities are conducted in four major areas: Advanced Passive Solar Materials Research, Collector Technology Research, Cooling Systems Research, and Systems Analysis and Applications Research.

Advanced Passive Solar Materials Research. This activity area includes work on new aperture materials for controlling solar heat gains, and for enhancing the use of daylight for building interior lighting purposes. It also encompasses work on low-cost thermal storage materials that have high thermal storage capacity and can be integrated with conventional building elements, and work on materials and methods to transport thermal energy efficiently between any building exterior surface and the building interior by nonmechanical means.

Collector Technology Research. This activity area encompasses work on advanced low-to-medium temperature (up to 180°F useful operating temperature) flat plate collectors for water and space heating applications, and medium-to-high temperature (up to 400°F useful operating temperature) evacuated tube/concentrating collectors for space heating and cooling applications. The focus is on design innovations using new materials and fabrication techniques.

Cooling Systems Research. This activity area involves research on high performance dehumidifiers and chillers that can operate efficiently with the variable thermal outputs and delivery temperatures associated with solar collectors. It also includes work on advanced passive cooling techniques.

Systems Analysis and Applications Research. This activity area encompasses experimental testing, analysis, and evaluation of solar heating, cooling, and daylighting systems for residential and nonresidential buildings. This involves system integration studies, the development of design and analysis tools, and the establishment of overall cost, performance, and durability targets for various technology or system options.

This report is an account of research conducted in Systems Analysis and Applications concerning experimental testing of solar thermosyphons for residential hot water heating systems.

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ABSTRACT

The advantages of solar thermosyphons in terms of simplicity, reliability, and cost have long been recognized. Recent studies have also shown their thermal performance to be comparable with that of equivalent active systems. When pump power is considered, the energy savings of domestic hot water thermosyphons can be significantly superior to active systems. In spite of these advantages, use of solar thermosyphons in the United States is almost negligible compared to their widespread use in other countries. A major limitation to the use of thermosyphons in the United States is lack of effective, reliable freeze protection. One technique for reliable, passive freeze protection is to use a heat exchanger in the storage tank and a nonfreezing fluid in the collector. Previous analytical work indicates that the performance penalty for these systems with practical-sized heat exchangers may be small enough to make these systems economically feasible. A full-scale, residential-size test facility has been constructed for testing this concept and validating the theoretical models.

This paper describes results of testing comparing the performance of a horizontal tank with and without heat exchanger to a baseline case of a vertical tank without heat exchanger. An analytical expression for a "heat exchanger penalty factor" for those systems is derived and compared with the experimental results.

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NOMENCLATURE

A = surface area, m^2 (ft^2)

C_1 = heat exchanger penalty factor, Eq. (9)

C_2 = heat exchanger penalty factor, Eq. (10)

F' = plate efficiency factor

H = total power input to the collector absorber plate, kW (Btu/hr)

I = power input per unit area of collector = H/A_c, kW/m^2 ($Btu/hr\text{-}ft^2$)

q = heat transfer rate, kW (Btu/hr)

Q = total collector input to the loop, kWh (Btu)

t = time, hr

T = bulk average temperature, °C (°F)

$\overline{T}$ = time average temperature, °C (°F)

ΔT_t = tank operating temperature difference = $\overline{T}_t - \overline{T}_a$, °C (°F)

U = overall heat transfer coefficient, $kW/m^2\text{-}°C$ ($Btu/hr\text{-}ft^2\text{-}°F$)

η = efficiency, Eq. (12)

Subscripts

a = collector ambient

abs = collector absorber plate

c = collector fluid

exp = experimental

f = fluid

g = glycol

hx = heat exchanger

l = losses

o = base conditions

p = pipe; collector absorber plate

t = tank

w = water

1.0 INTRODUCTION

The simplicity and reliability of thermosyphons give them a significant advantage over active systems for domestic hot water heating applications. The thermal performance of thermosyphons has been shown to be comparable with equivalent active systems [1], and when pump power costs are considered, their energy saving performance can be superior. Although these passive systems have been used extensively in other countries, active systems have found much more acceptance than thermosyphons in the United States. Application of passive systems in the United States has been limited by the following two major factors:

(1) Since the tank must be mounted above the collector in most cases, a vertical storage tank presents a structural and aesthetic problem.

(2) Freezing of the low mass collector is a serious problem in almost all United States climates.

The first problem can be partially overcome by mounting the tank in a horizontal position, making it easier to locate inside an attic and more effectively distributing the structural loads. Studies by Young et al. [2-4] indicate that performance of traditional thermosyphons may not be seriously affected when horizontal tanks are used. Their results show that while axial stratification is negligible, vertical stratification can still be significant in horizontal tanks subjected to typical thermosyphon flowrates if a supply water diffuser is used. Other experimental results from Huang [5] indicate that the performance of a low-resistance thermosyphon is insensitive to tank elevation. This result is also consistent with the theoretical predictions of Mertol et al. [6] for systems with heat exchangers. Therefore, lowering the tank relative to the collector to mitigate the structural/aesthetic problem may not seriously affect performance, as long as reverse flow at night can be suppressed.

One way to overcome the freeze problem is to use a heat exchanger in (or on) the storage tank and a nonfreezing fluid in the collector loop. This method of freeze protection has been used extensively for active systems, but its application to thermosyphons is not well understood. For cold climates, this method has an advantage over others (such as solenoid activated drain down) in that it is safe from hard freeze conditions and completely passive; it therefore preserves the important attributes of simplicity and reliability inherent in the thermosyphon concept.

A bibliography of the numerous theoretical and experimental studies on solar thermosyphons contained in the literature, as well as summaries of the important findings of some of these studies, are provided in a previous paper by the authors [6]. The previous paper also presents results of computer simulations of a solar thermosyphon system with heat exchanger in the storage tank and propylene glycol working fluid in the collector loop. The results of that study indicate that the performance penalty for systems with heat exchangers of practical size (a size that can be used, constructed, and installed in a typical hot water system) may be small enough to make these systems economically feasible. Additional results show the effect on performance of different system parameters and indicate that the performance of these systems is relatively insensitive to tank stratification, tank elevation relative to the collector, and system resistance. Other studies of solar thermosyphons with heat exchangers at the storage tank using single phase fluids in the collector loop are not present in the literature.

The results of the work outlined above indicate that there may be significant potential for horizontal tank thermosyphons with heat exchangers using single-phase nonfreezing fluids in the collector loop. The present work addresses this problem and presents test results for one configuration of this type. This work is intended to provide a technical foundation for the concept, which could be used to foster implementation of these systems in cold climates.

2.0 CONCEPTUAL FRAMEWORK

The primary purpose of this experimental work was to ascertain the effect on thermosyphon performance of using a heat exchanger in the storage tank with propylene glycol in the collector loop. The approach for determining this effect was to compare collector performance between systems with and without heat exchangers. The theoretical basis for this comparison involves deriving an expression for steady state collector efficiency that includes a "heat exchanger penalty factor" (similar to that developed for active systems by DeWinter [7]). To derive such an expression for a closed loop thermosyphon, the following assumptions are made:

- Following the approach of Close [8], the average collector fluid temperature, $T_c = (T_{c_{out}} + T_{c_{in}})/2$, is assumed to be equal to the average heat exchanger fluid temperature, T_{hx}, at any time. (For the remainder of this derivation $T_c = T_{hx} = T_f$).

- Collector output is assumed to follow the Hottel-Whillier-Bliss (HWB) equation [9,10] over the operating range of interest:

$$q_c = F'(H - (UA)_c\,(T_f - T_a)) \tag{1}$$

  where H is the net heat input to the collector absorber plate, $(UA)_c$ is considered to be independent of temperature and U is the overall conductance between the plate and ambient air.

- Thermosyphon loop thermal capacitance is ignored.

- The logarithmic mean temperature difference for the heat exchanger is approximated by the difference between average collector loop temperature and bulk average tank temperature, so that heat transfer through the heat exchanger is given by

$$q_{hx} = (UA)_{hx} (T_f - T_t) \ . \tag{2}$$

- Pipe losses are approximated by

$$q_{pl} = (UA)_p (T_f - T_a) \ . \tag{3}$$

Since thermal capacitance is ignored, an overall energy balance for the thermosyphon loop (collector/heat exchanger/piping part of the system) can be written as

$$q_c - q_{pl} = q_{hx} \ . \tag{4}$$

Substitution of Eqs. (1-3) into Eq. (4) yields

$$F'(H - (UA)_c(T_f - T_a)) - (UA)_p(T_f - T_a) = (UA)_{hx}(T_f - T_t) \ . \tag{5}$$

The average fluid temperature, T_f, can be obtained from Eq. (5) as

$$T_f = \frac{T_a F'(UA)_c + (UA)_p T_a + T_t(UA)_{hx} + F'H}{(UA)_{hx} + (UA)_p + F'(UA)_c} \ . \tag{6}$$

Substitution of Eq. (6) into Eq. (1) yields an expression for the collector delivery rate as a function of the tank fluid to ambient temperature difference:

$$q_c = \frac{(UA)_{hx} + (UA)_p}{(UA)_{hx} + (UA)_p + F'(UA)_c} F'H - \frac{F'(UA)_c(UA)_{hx}}{(UA)_{hx} + (UA)_p + F'(UA)_c} (T_t - T_a) \tag{7}$$

Finally, the collector efficiency can be expressed as

$$\eta_c = \frac{q_c}{H} = F'C_1 - F'C_2(UA)_c \frac{T_t - T_a}{H} \tag{8}$$

where C_1 and C_2 are heat exchanger penalty factors given by

$$C_1 = \frac{(UA)_{hx}}{(UA)_{hx} + (UA)_p + F'(UA)_c} = C_2 + \frac{(UA)_p}{(UA)_{hx} + (UA)_p + F'(UA)_c} \tag{9}$$

and

$$C_2 = \frac{(UA)_{hx}}{(UA)_{hx} + (UA)_p + F'(UA)_c} \ . \tag{10}$$

Since $(UA)_p$ is generally small compared with $(UA)_{hx}$ (on the order of 1% for systems using heat exchangers of practical size and well insulated pipes), the last term on the right-hand-side of Eq. (9) can be dropped. Setting $C_1 = C_2 = C$ in Eq. (8) simplifies the relation for collector efficiency as follows:

$$\eta_c = CF' \left(1 - (UA)_c \frac{T_f - T_a}{H} \right) . \tag{11}$$

If no heat exchanger is used, $T_t = T_f$, $(UA)_{hx}$ is infinite, $C = 1$, and Eq. (11) reduces to the familiar HWB efficiency equation [9,10],

$$\eta_c = F' \left(1 - (UA)_c \frac{T_f - T_a}{H} \right) . \tag{12}$$

Now, assuming that quasi-steady state conditions are achieved in actual systems over an hour's time, then hourly average values can be substituted for the instantaneous variables in this equation (e.g., $T = \overline{T}$).

Equation (11) implies that to first order, the collector efficiency for a closed loop thermosyphon is dependent on the tank temperature operating point, $(\overline{T}_t - \overline{T}_a)/H$, and the "heat exchanger penalty factor," C. This result is analogous to DeWinter's formulation for active systems with double loop heat exchangers [7]. The advantage of this relatively simple expression for collector performance of thermosyphons with heat exchangers is in predicting long-term system performance. If the collector performance of these systems can be adequately represented by Eq. (11) over a wide range of operating conditions, then long-term system performance can be estimated by a relatively simple modification of existing traditional thermosyphon system performance algorithms [8,11,12].

3.0 EXPERIMENTAL FACILITY

3.1 *Experimental Apparatus*

The test apparatus (shown in Fig. 1) consisted of the following elements:

- Two solar collectors with a total net aperture area of 3.47 m^2 (37.36 ft^2) with selectively coated copper absorber plates, single-glazed covers, and with twenty-eight 0.025-m (1-in) wide by 1.83-m (72-in) long strip heaters attached to the back of the absorber plates.

- A cylindrical storage tank of 0.51-m (20-in) diameter and 1.52-m (60-in) length, insulated with a 0.1-m (4-in) thick layer of foam glass insulation.

- A tank support structure that allowed both vertical and horizontal orientation of the tank and permitted raising or lowering the tank relative to the collectors.

- Collector-to-tank piping made of 0.025 m (1 in) inside diameter silicone hose insulated with 0.018-m (0.75-in) thick elastomeric foam.

- A heat exchanger made up of a total of eight copper tubes, each with an outside diameter of 0.025 m (1 in), wall thickness of 0.003 m (0.13 in), and length of 1.52 m (5.0 ft), running parallel to the axis of the tank. Two manifolds at each end of the tank were connected to the heat exchanger tubes with a short length of flexible silicone hose; by clamping this piece of hose, the flow in any of the tubes could be stopped (i.e., the operative heat transfer area could be changed).

The collectors were mounted inside a 3-m (10-ft) by 9-m (30-ft) trailer where all of the electrical and data recording equipment was located. The trailer was conditioned with a 3.5 kW (1.5 Ton) air conditioner that limited the maximum temperature in the trailer to 23-24 °C (72-75 °F); the minimum temperature was uncontrolled. The tank and its support structure were located outside of the trailer; they were shielded from solar radiation by a canvas canopy, thereby allowing the use of outdoor air temperature as the sole indicator of the thermal environment around the tank.

Other elements used in the experiment included a small pump for mixing the storage tank between tests, a bucket connected to the tank to permit tank water expansion, an expansion tank for heat exchanger fluid expansion, and air vents located at high points of the system. The water stored in the tank was drawn from a city water supply and was replaced whenever a low initial tank temperature was desired. The heat exchanger fluid was a 60% solution of propylene-glycol and water.

3.2 *Instrumentation and Control Equipment*

The temperatures throughout the system were measured with type T thermocouples. Two types of thermocouples were used. All thermocouples, except for those used to measure collector fluid temperature, were made from 0.635-mm (0.025-in) diameter wire. Collector and heat exchanger inlet and outlet temperatures were measured by sheathed thermocouples (fabricated from thin stainless steel tubing and the above thermocouple wire) that were inserted into the fluid stream through tight-sealing fittings. All thermocouple wires were connected to a 60-channel data acquisition system with output to magnetic tape. The data acquisition system and samples of both kinds of thermocouple wire (as well as both types of thermocouples) were calibrated to determine systematic errors in the measuring system and to estimate random errors in the temperature readings. The calibration results indicate that temperature measurements were accurate to ±5 °F.

Figure 2 is a schematic diagram of the tank cross section showing the location of heat exchanger tubes and the thermocouple grids used inside the horizontal tank. The grids were designed to allow an accurate calculation of the weighted average tank temperature and to allow determination of the tank temperature distribution. Three such grids, with a total of 44 thermocouples, were spaced evenly along the horizontal axis to assess possible longitudinal temperature variations. When the storage unit was set vertically, the average tank temperature was measured using six equally spaced thermocouples mounted on a rod located near the center line of the tank. Other temperature measurements included the ambient temperature inside the trailer near the collectors, the ambient temperature outside of the trailer near the tank, and the collector absorber plate temperature.

Power to the collector strip heaters was controlled by a power conditioning system that consisted of a burst-firing, SCR power controller and a microprocessor-based analog/events programmer. The power conditioning system was programmed to provide a half-period sine function power input profile to the collectors to simulate a solar profile. A watthour meter was used to read the total energy input to the collectors during testing.

4.0 EXPERIMENTAL METHODOLOGY

4.1 *Preliminary Testing*

Preliminary and ancillary tests included calibrations of the SCR controller and thermocouples and measurement of tank and collector *UA* values. The tank *UA* value was found by measuring the tank temperature decay over a range of operating temperatures. The average value determined was 4.6 W/°C (8.8 Btu/hr-°F) and was essentially independent of temperature over the range of 21-66°C (70-150 °F). This compared with a theoretical value of 3.4 W/°C (6.5 Btu/hr-°F), which did not account for tank fittings. The uncertainty in the experimental value was ±0.246 W/°C (±0.466 Btu/hr-°F) to 95% confidence limit.

The collector *UA* value was found at stagnation conditions by closing off the inlet and the outlet collector piping and then measuring the equilibrium plate temperature at a given power input [13-15]. (Six thermocouples were equally spaced along the length of the collector near the center line and were attached at the fin root with sheet metal screws.) From these tests, a correlation between collector *UA* and T_{abs} (Fig. A1) was found. From further tests, a correlation between T_{abs} and T_f (Fig. A2) was found. Using these two correlations, collector *UA* could be determined from T_f (see Appendix A).

4.2 *Performance Testing*

Three configurations were studied:

(1) Vertical tank without heat exchanger.

(2) Horizontal tank without heat exchanger.

(3) Horizontal tank with heat exchanger.

In each case the bottom of the tank was approximately .6 m (2.0 ft) above the top of the collector. For the last configuration the heat exchanger consisted of straight tubes passing through the tank near the bottom (cf. Figs. 1 and 2) and was constructed so that the heat transfer area could be varied by closing off tubes. None of the tests were conducted with water drawn from the tank to simulate a hot water load. Every attempt was made to keep the systems as close to identical as possible for each configuration. However, there were some unavoidable differences in piping length between the vertical and horizontal cases.

The basic approach was to test each configuration under a wide variety of operating conditions. This was accomplished by conducting all-day runs with various tank starting temperatures with simulated solar input profiles of different peak magnitudes and durations. A large number of runs was made for each configuration studied. Each run was started with the tank filled with either fresh cool water or hot water from a previous run. The tank was mixed prior to each run to remove any stratification. Two basic half sine, wave power input profiles were used, 9 hours and 14.5 hours long, corresponding to winter and summer days, respectively. The magnitude of the sine wave peaks was varied between 1150-2300 W (3925-7850 Btu/hr) during a series of runs for each configuration. Once a run was started, the control and data gathering was completely automatic.

The collector flowrate was not measured due to the unavailability of a suitable low-flow, low-resistance, flow-measuring device. All heat transfer rates were therefore calculated from temperature measurements; four 15-minute data scans were averaged to determine the performance parameter. The total energy input was measured with a watthour meter and the final value compared to the integrated value based on the SCR calibration curve.

5.0 RESULTS AND DISCUSSION

Typical results for a high-temperature and low-temperature test are shown in Figs. 3 and 4, respectively. Figures 3 and 4 also show the lag between collector temperature and tank temperature relative to the input profile. The lag in collector loop temperature is due primarily to tank water thermal capacitance but is also influenced by collector loop thermal capacitance. These figures also show that the assumption that $T_c = T_{hx}$, used in the theoretical formulation in Section 2, is valid for the majority of operating points. (T_c refers to the collector fluid temperature.)

The determination of the performance penalty from utilizing a heat exchanger in the storage tank was based on correlating measured instantaneous collector efficiency with tank operating parameter (see Section 2). For this analysis each test point within a run was derived from an hourly average heat balance. Examination of the test results indicated that points during periods of collector stagnation or during transient flow conditions at the beginning and end of a test, where the assumption of quasi-steady-state behavior was considered most questionable, should not be included. During these periods large errors are introduced by assuming that collector output was equal to the sum of tank gain, tank losses, and pipe losses. Consequently, a conservative approach to rejecting test points was adopted so that all data points used for the correlation would correspond as closely as possible to quasi-steady-state behavior. The criterion used was to reject all points within the first three and last two hours of a test run and any other points where the tank energy gain was not greater than the sum of tank and pipe losses.

Figure 5 shows the final data set for the eight-tube case. The scatter and operating range shown by this figure is characteristic of results for the other configurations analyzed. The line shown was found from a nonweighted, least-squares fit to the data.

The accuracy of the linear fit is supported by comparing the total daily collected energy as calculated from the linear curve fit (i.e., calculated from a daily summation of

15-minute collector energy outputs as determined from the curve fit.), with that measured by the watt hour meter for an individual test run. As indicated in Table 1, the differences are less than 4% for any of the configurations studied. These results indicate that the collector performance of thermosyphons with heat exchangers can be reasonably well approximated by a linear relationship between collector efficiency and tank operating parameter.

Figure 6 shows a plot of fitted equations for the major configurations tested. The figure indicates that there is no apparent difference between the vertical and horizontal cases without heat exchangers. The figure also indicates that there is a substantial performance penalty associated with the use of a heat exchanger in the horizontal tank configuration. To quantify this penalty, an experimental "heat exchanger penalty factor," C_{exp}, can be derived by taking a ratio of Eqs. (11 and 12):

$$\frac{\eta_i}{\eta_o} = C_{\mathrm{exp}} \left(\frac{F'_g}{F'_w} \right) \frac{1-(UA)_{c_i} \dfrac{\overline{T}_t - \overline{T}_a}{I}}{1-(UA)_{c_o} \dfrac{\overline{T}_t - \overline{T}_a}{I}} \quad . \tag{13}$$

where $(UA)_{c_i}$ is the collector UA corresponding to a particular heat exchanger curve and $(UA)_{c_o}$ is the collector UA corresponding to the no heat exchanger case. This equation allows the heat exchanger penalty factor to be determined from a comparison of the test results for the no heat exchanger cases with those for the heat exchanger case. To correct for the difference in fluids, the ratio $F'_g/F'_w = 0.988$ was calculated by theoretical equations given in Ref. [9]. The results of these calculations are shown as a solid line in Fig. 7.

Although in actuality the difference in $(UA)_c$ (due to differences in operating temperature of the collector) between the no heat exchanger and a heat exchanger case at any tank operating point varies with the operating parameter, C_{exp} was calculated by using a constant $(UA)_c$ for each configuration equal to the average value derived from all the data points for that configuration. (These average $(UA)_c$ values were determined from average collector temperature data as outlined in Appendix A; the resulting values are shown in the table in Fig. 7). The error introduced in C_{exp} by making this assumption was estimated to be less than 1%.

The data points shown on Fig. 7 were derived from a calculation of the penalty factor using Eq. (9) along with instantaneous experimental $(UA)_{hx}$ values and the average $(UA)_c$ values from the table in Fig. 7; these points serve as a semi-independent check on the consistency of the data and the determination of the heat exchanger penalty factor. The curve indicates some variation in heat exchanger performance with operating point. This variation is less than the uncertainty in the data used to determine this curve. Assuming a constant value for C would not appear to introduce significant error when using the heat exchanger penalty factor for long-term performance predictions.

These results show that the instantaneous collector performance for the system studied can be adequately represented by the simplified theoretical formulation derived in Section 2 and expressed by Eqs. (9) and (12). It is possible that these results would be applicable over a fairly broad range of operating conditions other than those tested, without introducing a significant error in long-term performance predictions; i.e., the basic assumption that $T_c = T_{hx}$ would be valid for the hours of primary energy collection for most systems, and the operating conditions covered by the testing were representative of those of typical practical systems.

Figure 8 shows the expected variation in C for different ratios of $(UA)_{hx}/(UA)_c$ as calculated by Eq. (9). The portion of the curve of interest to practical system design is shown as a solid line. Over this portion of the curve, variations in collector and piping heat loss coefficients are relatively unimportant. This curve gives an indication of how large the heat exchanger capacity must be to limit the performance penalty for the majority of daily operating points. For example, to limit the performance penalty to less than 10%, the ratio must be greater than 8. For the simple heat exchanger studied, this corresponds to a heat-exchanger-to-collector-area ratio of roughly 0.25-0.35. It should be pointed out that this curve gives only an indication of the effect of heat exchanger capacity on instantaneous collector performance; it does not show the effect on long-term performance or for which range of operating conditions the $(UA)_{hx}$ needs to be selected to keep the long-term performance penalty low.

Finally, Fig. 9 compares the tank temperature profiles at different times during a test. The broken lines indicate the profiles without a heat exchanger. The solid lines are the profiles for the case with a heat exchanger; these curves show that the significant stratification observed for the horizontal tank without heat exchanger is essentially eliminated. This difference in tank stratification is the result of two major factors:

- Due to the inherent limitations of the heat exchanger configuration, hot water is not introduced directly to the top of the tank.

- Hot water rising from the heat exchanger tubes mixes with the adjacent cool water before it can accumulate at the top of the tank.

6.0 CONCLUSIONS

The following conclusions summarize theoretical and experimental results presented in this paper:

- The experimental results validate the simplified theoretical model assumption that average collector fluid temperature is equal to average heat exchanger fluid temperature over the operating conditions of major importance to daily energy collection. Provided that this assumption were valid for other climatic and system sizing parameters, then the general conclusions of this paper could be generalized to a wide variety of other design and operating conditions.

- The results suggest that, to the first order, the degradation in collector performance due to utilizing a heat exchanger at the storage tank in thermosyphon systems can be accounted for by a "heat exchanger penalty factor." Moreover, if the heat exchanger $(UA)_{hx}$ is known or can be reasonably estimated, this penalty factor can be determined by a relatively simple expression given by Eq. (9).

- The results of the experiments with a simple, bare tube heat exchanger indicate that relatively large heat transfer areas may be required to limit degradation in instantaneous performance of these systems to less than 10% over typical operating ranges.

- The data indicate that for bare horizontal tubes installed at the bottom of a horizontal tank, very little tank stratification exists for the entire daily operation of the system. This suggests that loss of the benefits of tank stratification may be one of the consequences of utilizing some types of heat exchangers in thermosyphon systems. The magnitude of this effect on system performance, however, has not been determined.

- The performance of thermosyphons with horizontal tanks without heat exchangers can be comparable to that of vertical tanks without heat exchangers.

The above observations and conclusions should be considered with the following qualifications in mind:

- The experiments were conducted for only one system size and configuration (except for differences in tank orientation) and over a limited range of operating conditions.

- The experimental apparatus did not allow testing over operating points typical of cold weather operating conditions.

- The tank insulation was relatively poor, so tank losses were high, causing the system to stagnate at fairly high collector operating efficiencies.

- Data analysis was limited by the unavailability of a flowmeter and therefore the necessity to determine collector output from tank temperature rise and calculated heat losses. This limited the number of useful data points obtained during a test run, contributed to large scatter caused by thermal capacitance, and increased the uncertainty in the experimental results.

- No draw was used, so the system operation was not typical of actual operating conditions. This could have a significant impact on heat exchanger performance and tank stratification.

- Only one very simple heat exchanger configuration was tested. The results cannot be considered representative of optimal attainable performance for thermosyphon systems with all types of heat exchangers. However, the type of heat exchanger used is typical of at least one important class of heat exchangers that could be used in thermosyphons; the results provide considerable insight into the effect on performance of this class. Extension of the methodology that we have presented would lead to similar insights for other configurations.

- The ambient conditions at the collector and tank were essentially uncontrolled, restricting the opportunity for direct measurement of performance differences under identical operating conditions.

- The heat exchanger (UA) to be used in Eq. (9) was defined in terms of the difference between average temperatures rather than the logarithmic mean temperature difference. Theoretical U_{hx} values, therefore, must be adjusted for this change in these temperature differences before being used in Eq. (9). If experimental U_{hx} values are determined, they should likewise be determined from the difference of averages or adjusted if determined with logarithmic mean temperature difference.

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APPENDIX A:
COLLECTOR UA DETERMINATION

The collector (UA) used in the analysis of heat exchanger penalty was derived from ancillary testing that resulted in a correlation between collector UA and average collector fluid temperature, $\overline{T}_c$. To arrive at this correlation, two series of tests were conducted. First, the system was operated with a pump, a low-flow (but high resistance) flow meter (Kent Mini-Major) and thermocouples attached to the absorber plate fin root. Data were collected over a wide range of typical flow rates and temperatures to correlate plate temperature with collector average temperature; Fig. A1 shows these results. In the second series, stagnation tests were conducted on the collector using different power inputs while measuring absorber plate temperatures. Since collector input and losses are equal at this condition, the collector heat loss coefficient can be determined as a function of plate temperature. This correlation is shown in Fig. A2.

TABLE 1. Total Daily Collector Output for Horizontal Tank

Configuration	Test	Experimental Q kWh	Experimental Q kBtu	Predicted Q kWh	Predicted Q kBtu	Difference %
No Heat Exchanger	H0/A03L	10.00	34.13	10.14	34.60	1.4
	H0/A05S	9.85	33.63	10.25	34.99	4.0
Heat Exchanger (2 Tubes)	H2/A20S	4.83	16.48	4.99	17.02	3.2
	H2/A21S	6.37	21.72	6.35	21.68	-0.2
	H2/A22L	6.51	22.23	6.48	22.12	-0.4
Heat Exchanger (8 Tubes)	H8/A10L	6.08	20.76	6.24	21.31	2.6
	H8/A02L	8.54	29.16	8.50	29.01	-0.5
	H8/A13L	11.34	38.71	11.39	38.89	0.5

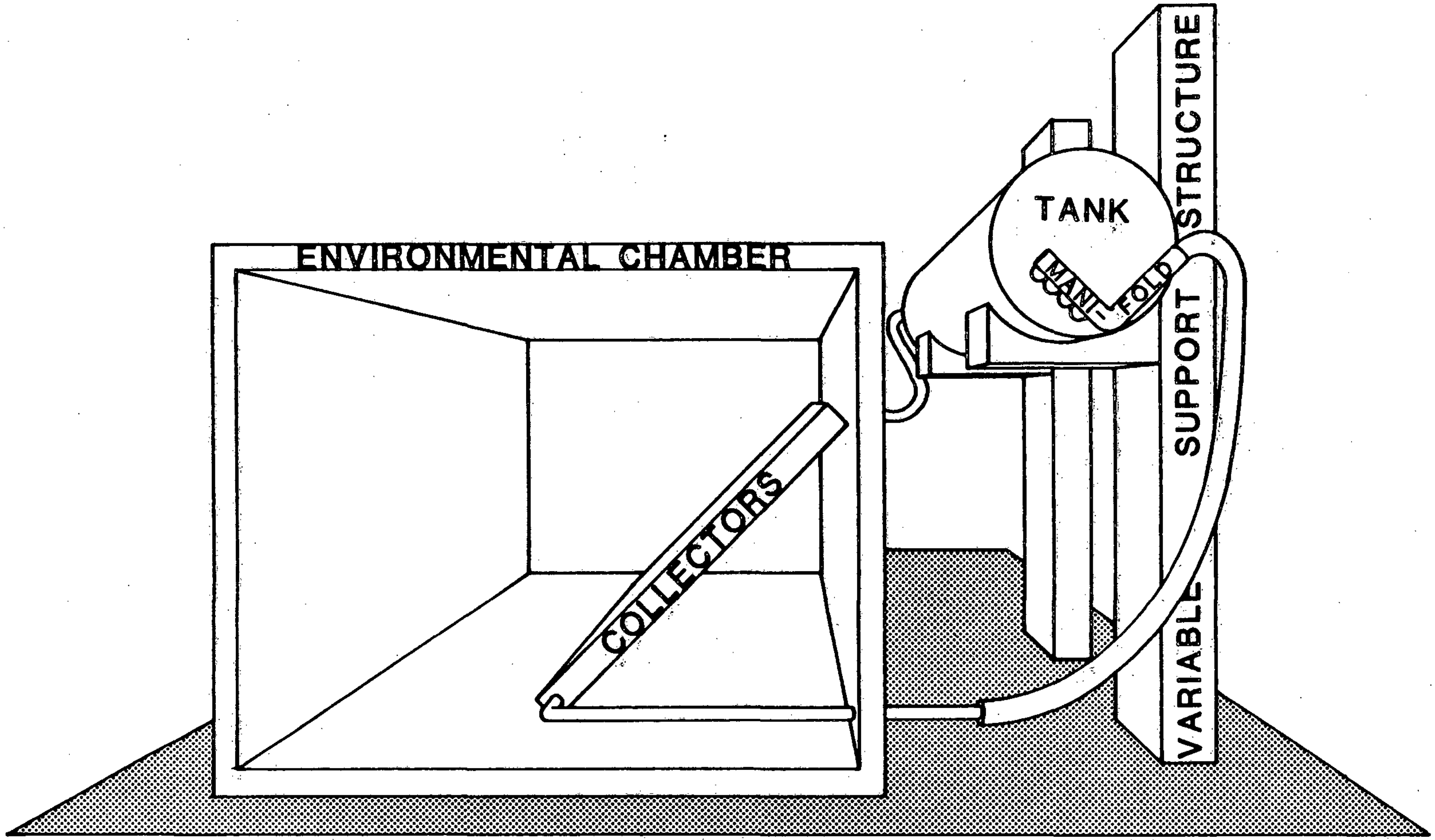

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Fig. 1: Schematic of experimental thermosyphon water heater.

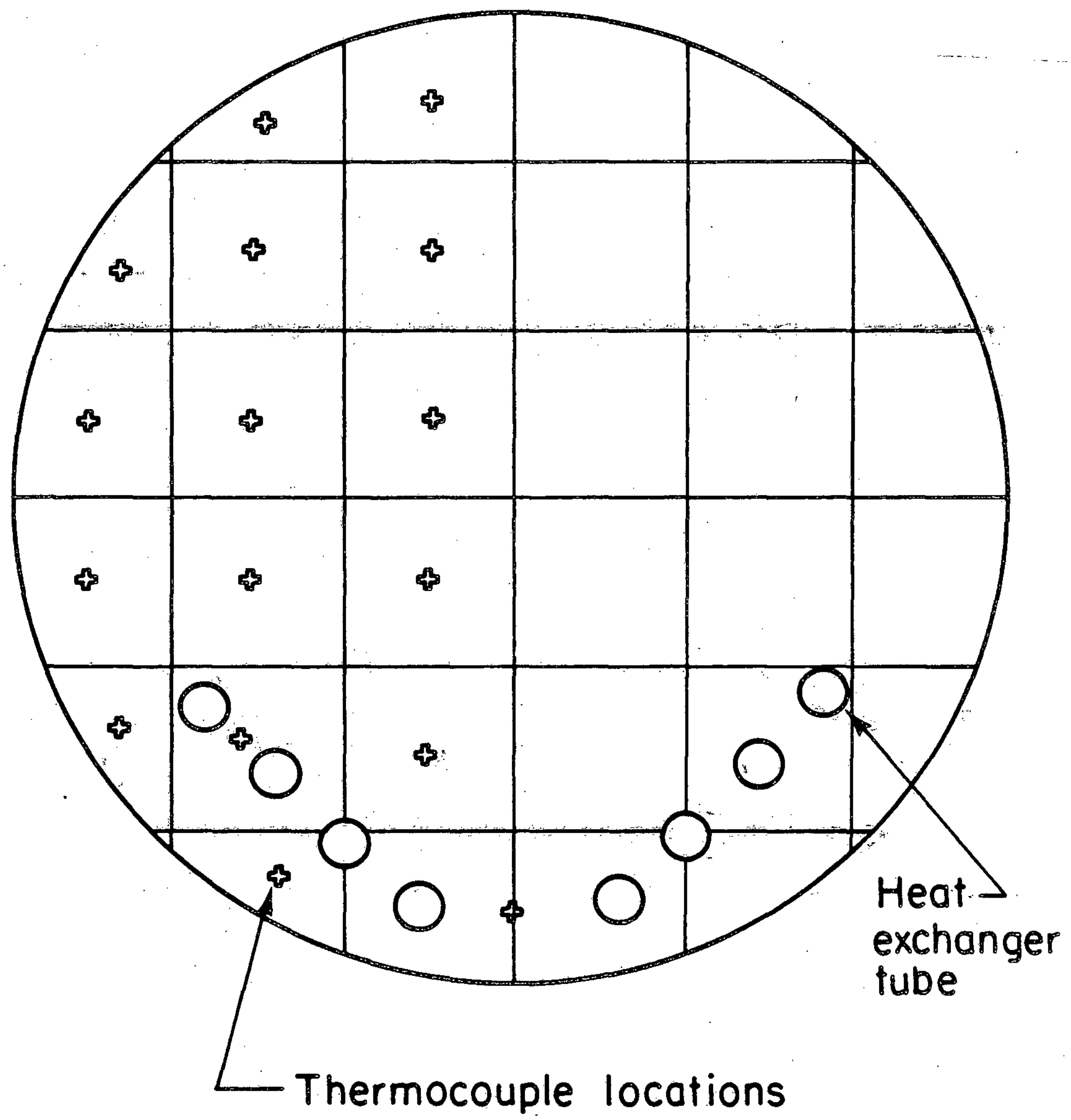

Fig. 2: Cross section of storage tank showing locations of thermocouples and heat exchanger tubes.

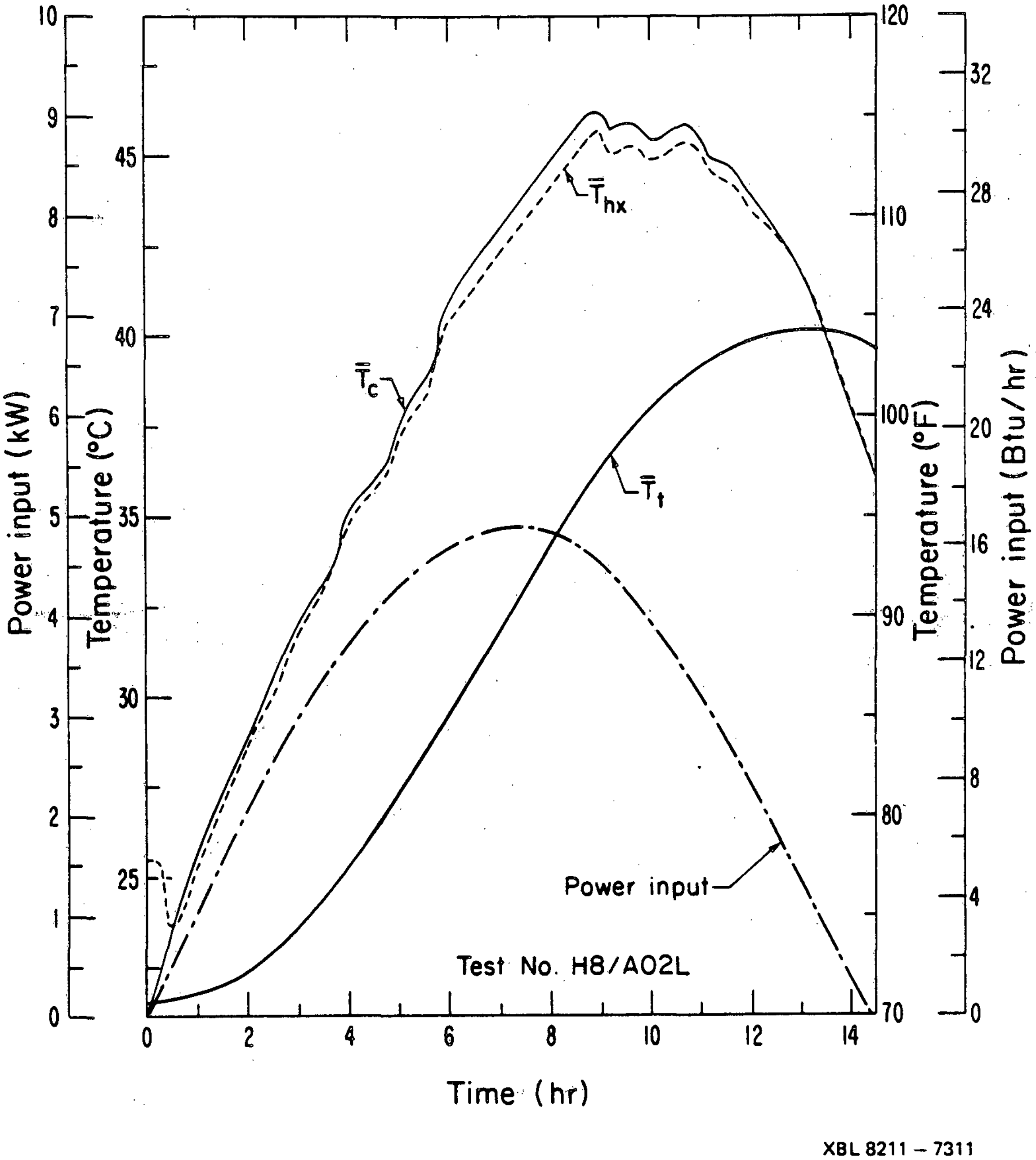

Fig. 3: Temperature profiles and power input for a typical high-temperature test.

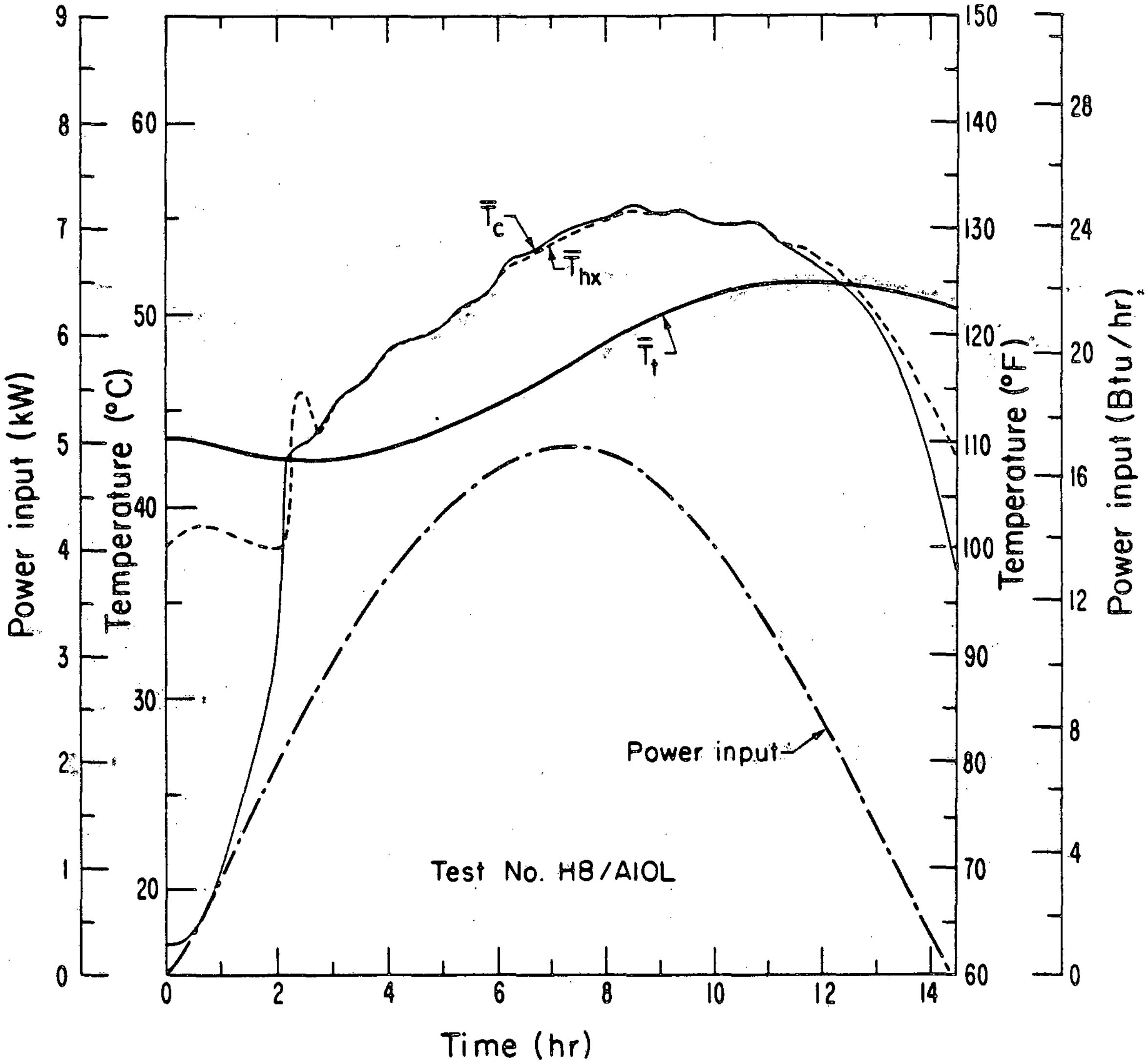

Fig. 4: Temperature profiles and power input for a typical low-temperature test.

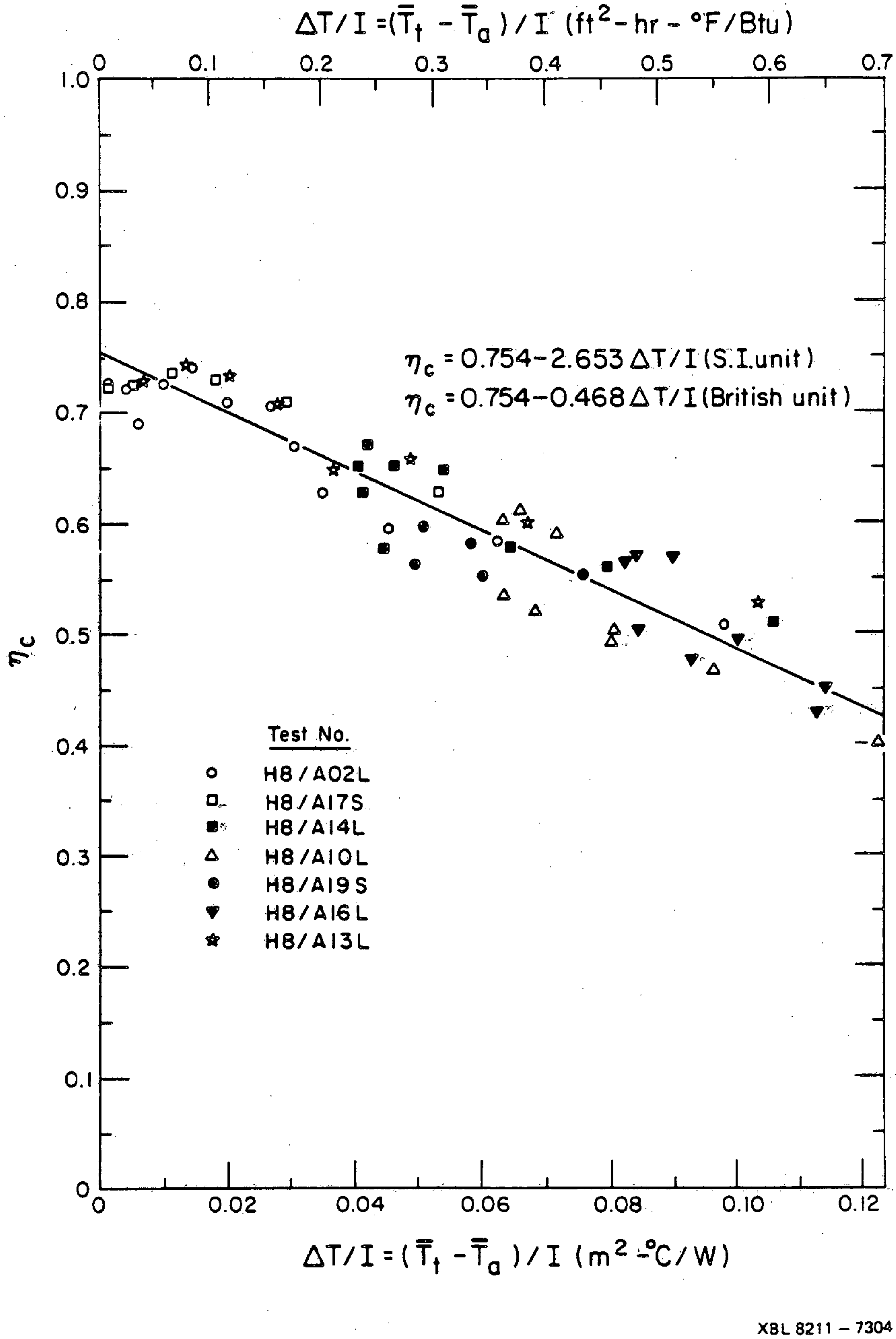

Fig. 5: Data point set and curve fit showing variation of collector efficiency with tank operating parameter for 8-tube tests.

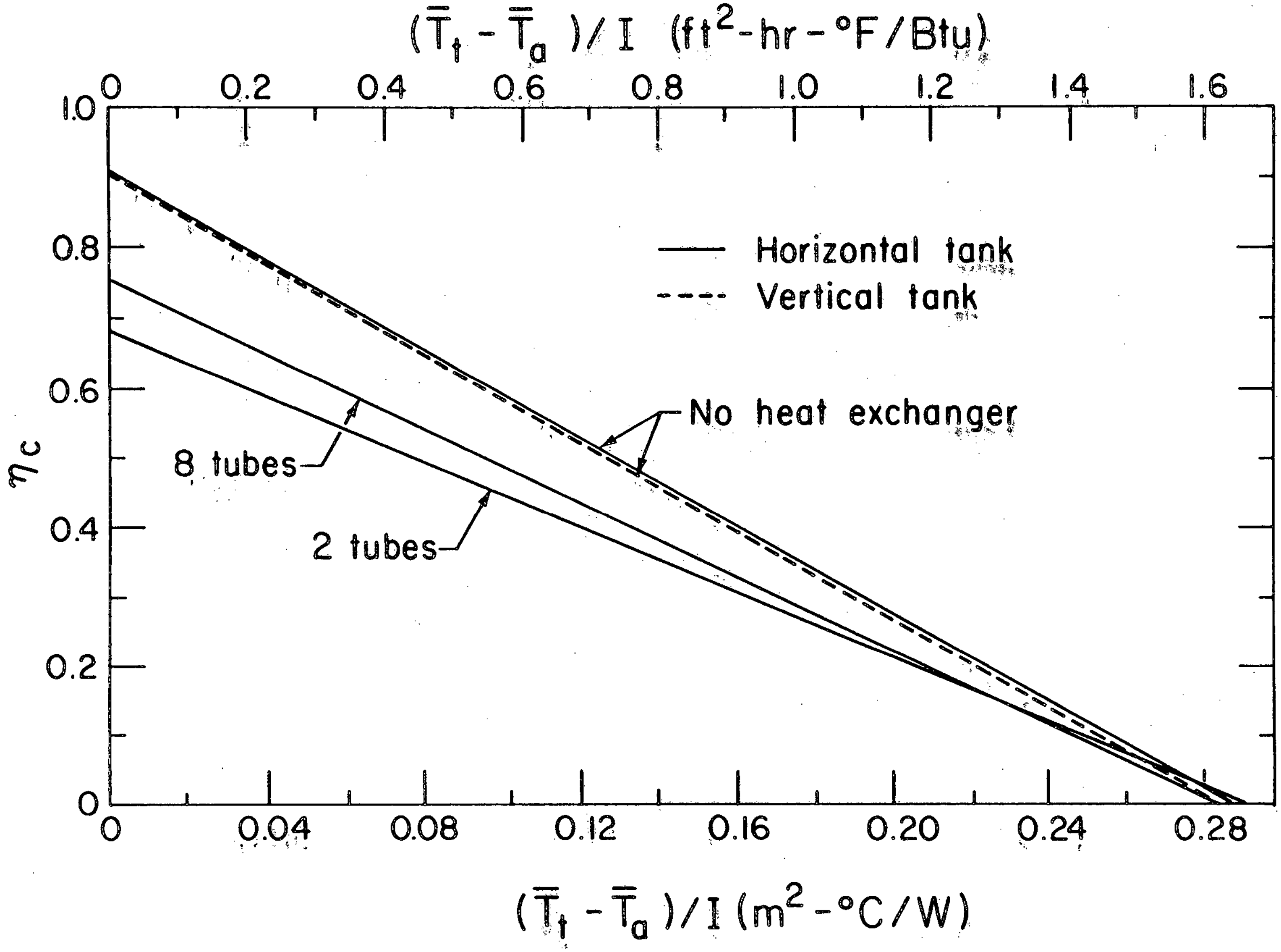

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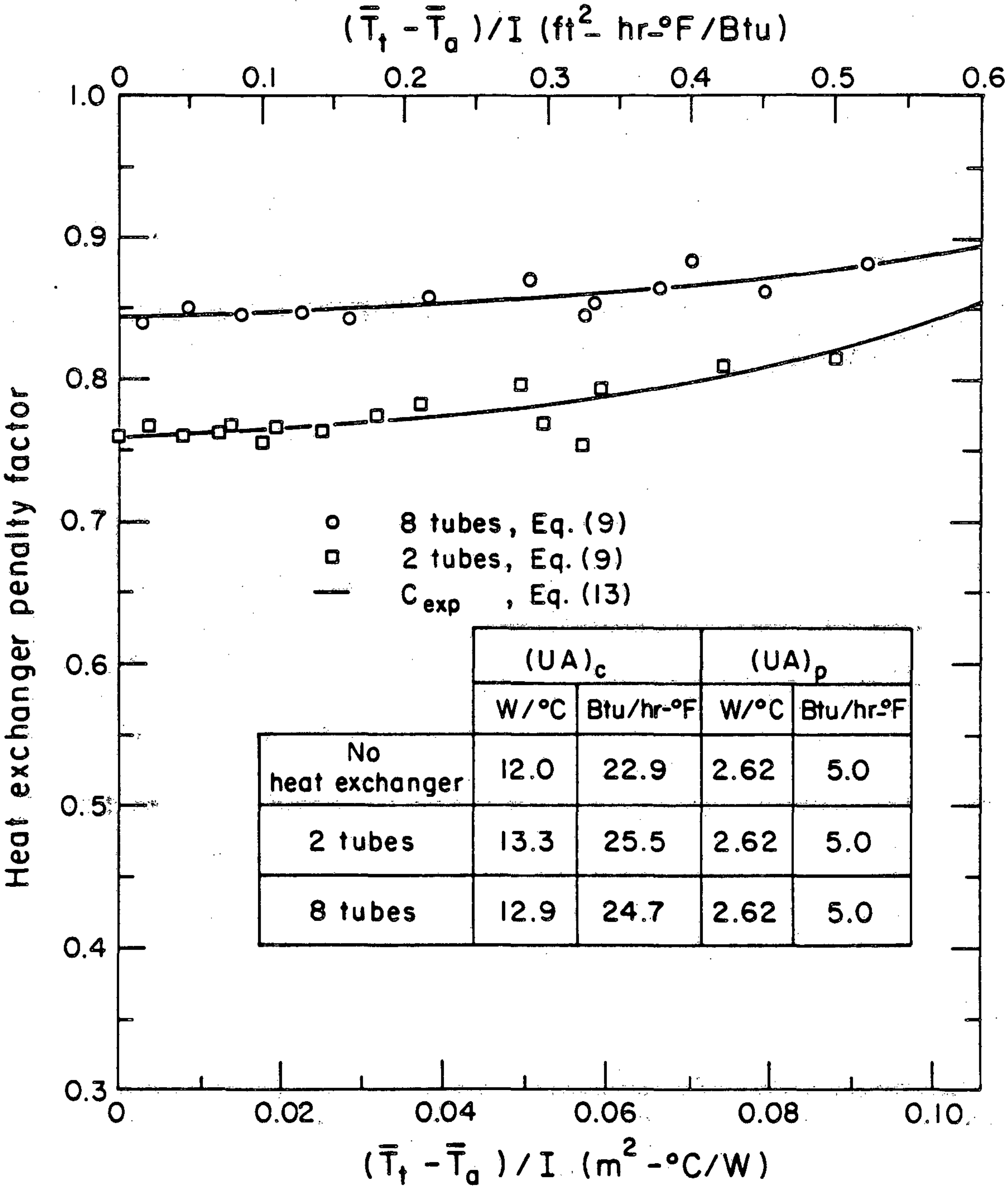
Fig. 6: Comparison of collector efficiency curve fits for various system configurations.



	$(UA)_c$		$(UA)_p$	
	W/°C	Btu/hr-°F	W/°C	Btu/hr-°F
No heat exchanger	12.0	22.9	2.62	5.0
2 tubes	13.3	25.5	2.62	5.0
8 tubes	12.9	24.7	2.62	5.0

XBL 8211 – 7306

Fig. 7: Variation of heat exchanger penalty factor with tank operating parameter.

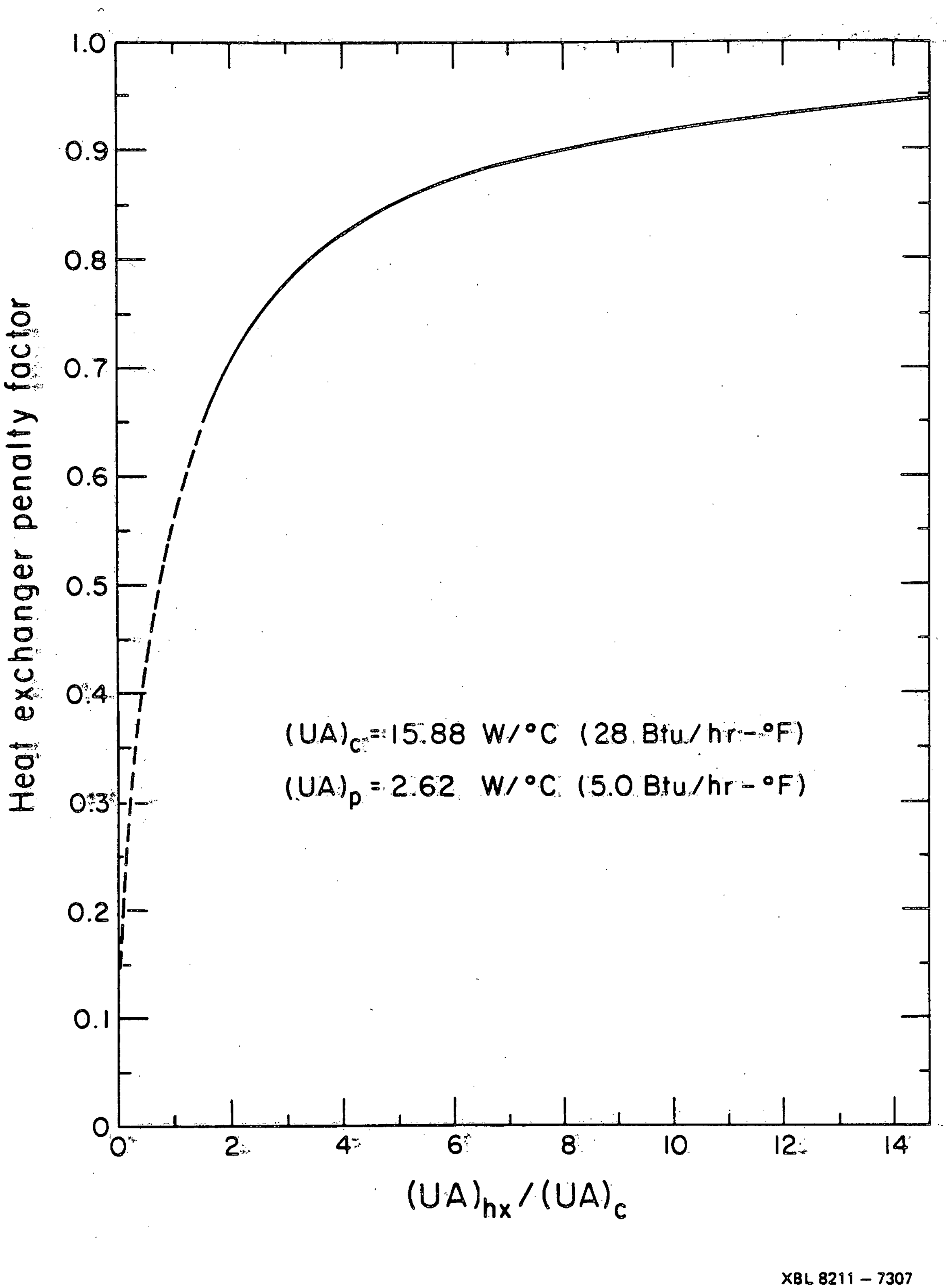


Fig. 8: Variation of heat exchanger penalty factor with heat exchanger capacity ratio.

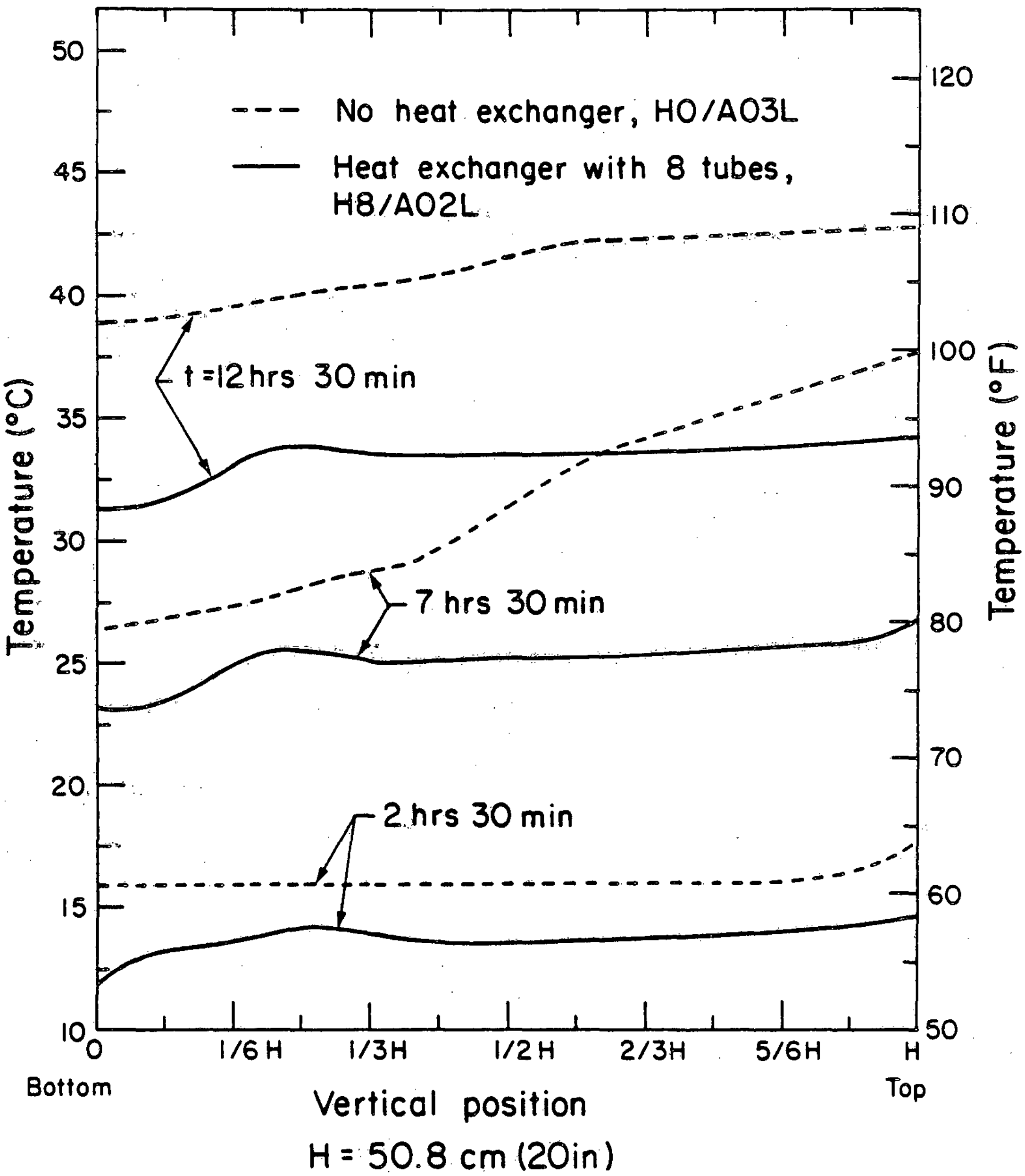
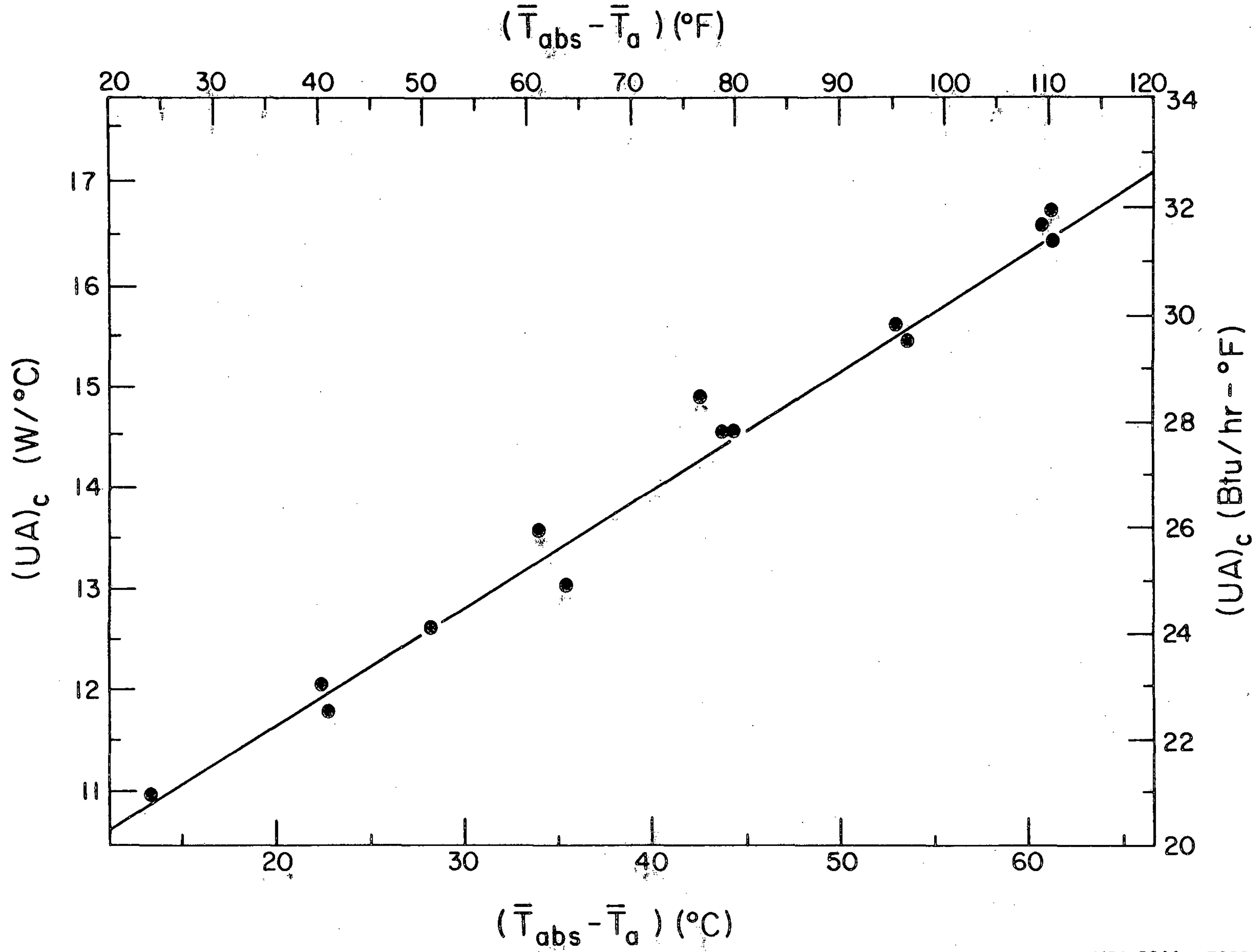


Fig. 9: Temperature stratification in the horizontal tank.



XBL 8211 - 7328

Fig. A1: Variation of collector overall loss coefficient with the difference between absorber plates and ambient temperatures.

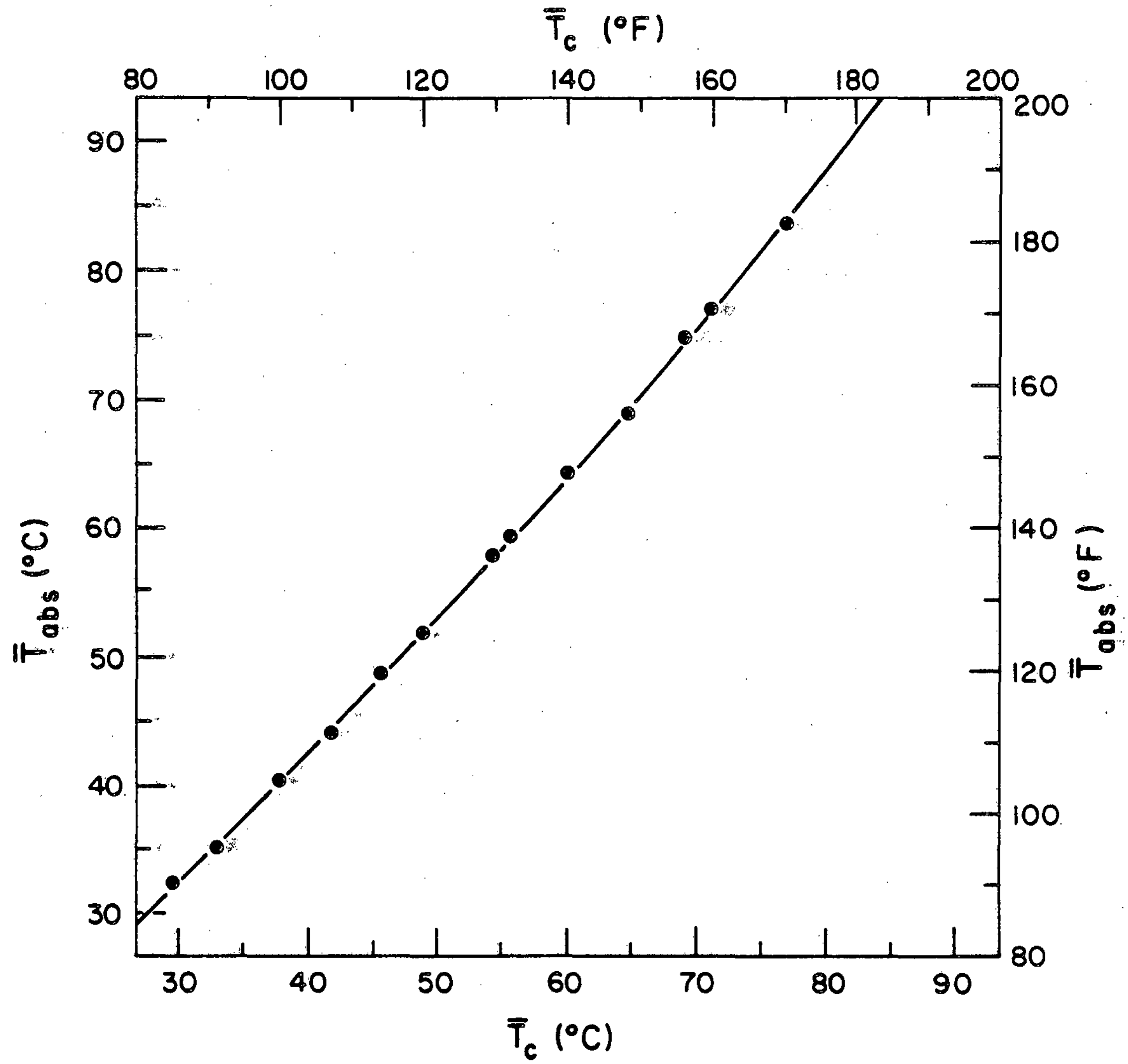


Fig. A2: Variation of the average absorber plate temperature with collector average fluid temperature.

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