Experimental Investigation of Nighttime Losses from ICS Solar Domestic Hot Water Systems

by

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(ABSTRACT)

The nighttime losses from an integral collector storage (ICS) system were investigated. The significance of the sky temperature, wind speed, and ambient temperature on the losses were examined. Outdoor data was taken on several nights to characterize the thermal performance of an ICS system under various environmental conditions. Indoor tests were then performed under an artificial "nighttime sky" environment, with a simulated wind, in an attempt to duplicate the heat losses which occurred outdoors.

The standard rating procedure which specifies the conditions for the heat loss tests for ICS systems was analyzed to see how well it characterizes the collector performance at night. Experimental results indicate a synergistic effect between the sky temperature and wind speed. The effects of wind on the losses from the ICS system overshadow the effects of small changes in sky temperature, but larger changes of sky temperature, with a constant wind speed, have a pronounced effect.

It is recommended that both of these parameters be taken into account in heat loss tests in standard rating procedures. Indoor tests can duplicate outdoor heat loss results within 8 per cent. The minimum re-

quirement for SRCC rating tests should be to monitor, record, and report the sky temperature.

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NOMENCLATURE

A_c	Collector aperture area, m^2
C_e	Effective specific heat of the tank and fluid, kJ/(kg C)
d	Cover plate spacing, m
E_s	Blackbody emissive power of the sky, W/m^2
F_{mn}	Radiation shape factor from surface m to surface n
Gr	Grashof number, dimensionless
h	Convective heat transfer coefficient, $W/(m^2\ C)$
h_w	Forced convection coefficient due to wind, $W/(m^2\ C)$
h_{rg}	Free convection coefficient between the reflector and the air in the enclosure, $W/(m^2\ C)$
h_{tg}	Free convection coefficient between the tank and the air in the enclosure, $W/(m^2\ C)$
h_{1g}	Free convection coefficient between cover 1 and the air in the enclosure, $W/(m^2\ C)$
h_{12}	Free convection coefficient between covers 1 and 2, $W/(m^2\ C)$
h_{23}	Free convection coefficient between covers 2 and 3, $W/(m^2\ C)$
J_c	Longwave diffuse radiosity leaving the bottom of cover 1, W/m^2
J_{c1}	Longwave diffuse radiosity leaving the top of cover 1, W/m^2
J_{c2}	Longwave diffuse radiosity leaving the bottom of cover 2, W/m^2
J_{c3}	Longwave diffuse radiosity leaving the top of cover 2, W/m^2
J_{c4}	Longwave diffuse radiosity leaving the bottom of cover 3, W/m^2

J_s	Longwave diffuse radiosity leaving the top of cover 3, W/m^2
J_r	Longwave diffuse radiosity leaving the reflector, W/m^2
J_t	Longwave diffuse radiosity leaving the tank, W/m^2
k	Thermal conductivity of air, W/(m C)
m_e	Effective mass of the tank and fluid, kg
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
q_{loss}	Energy lost by the tank fluid through convection and longwave diffuse radiation, W
Ra	Rayleigh number, dimensionless
R	Heat loss parameter, dimensionless
t	Time, s
T_a	Temperature of the ambient air, K
T_{act}	Temperature measured with ASTM thermometer, K
T_{bot}	Temperature of the fluid in the tank as measured by a thermocouple in the inlet connection, K
T_e	Effective sink temperature, K
T_g	Temperature of the air within the enclosure, K
T_{meas}	Temperature measured with a thermocouple and recorded by the system, K
T_r	Temperature of the reflector, K
T_s	Sky temperature, K
T_t	Temperature of the fluid in the tank, K
$\overline{T}_t$	Average hourly temperature of the fluid in the tank, K
T_{top}	Temperature of the fluid in the tank as measured by a thermocouple in the discharge connection, K

T_0	Initial Temperature of the fluid in the tank, K
T_1	Temperature of cover 1, K
T_2	Temperature of cover 2, K
T_3	Temperature of cover 3, K
$T_{w,avg}$	Weighted average of the fluid temperature in tank, K
T	Temperature, K
U_L	Overall heat loss coefficient, $W/(m^2\ C)$
U_{ra}	Loss coefficient through the reflector to the ambient air, $W/(m^2\ C)$
V_w	Wind velocity, m/s

Greek Symbols

ε	Emittance
τ	Transmittance
ρ	Reflectance
σ	Stefan-Boltzmann constant, $W/(m^2\ K^4)$
β	Collector slope, degrees
ζ	Arbitrary time interval
θ	Dimensionless temperature variable
θ_0	Dimensionless initial temperature
θ_f	Dimensionless final temperature

Subscripts

a	Ambient air

bot	Bottom of the tank
e	Effective
f	Final
g	Air within the collector
in	Indoor
out	Outdoor
r	Reflector
s	Sky
t	Tank
top	Top of the tank
w	Wind
0	Initial
1	Cover 1
2	Cover 2
3	Cover 3

1.0 INTRODUCTION

Integral collector storage (ICS) units are passive solar domestic hot water heating systems which are constructed as a single unit with the collection and storage functions combined. In the literature these units are also commonly referred to by several names including breadbox collectors, batch heaters, and integrated passive heaters. The collector is directly exposed to solar radiation during the day and may also be exposed to the nighttime sky during the evening. This latter situation results in nocturnal cooling which is the major concern of this investigation.

The study of nighttime losses from ICS systems has received little attention. Most theoretical analyses deal only with convective losses to ambient air. Radiative losses to the night sky are rarely accounted for directly and instead are assumed to be taken into account through the convective film coefficient. Some collector designs have incorporated a type of nighttime insulation which prevents losses from the storage tank, but generally most do not. This type of insulation is not favored by consumers because it is usually positioned manually and most people do not want to be bothered with daily placement and removal.

One reason why nighttime losses have been neglected is that in a conventional active system such losses generally do not occur. There is no circulation of fluid through the collector when it is not operating and the amount of fluid stored in the collector is either zero or negligible. The storage tank is instead located indoors and is well insulated.

With the increase in popularity of ICS systems over the past few years, it became necessary to develop a consistent and repeatable test standard. The federal government began requiring certification of collectors that were to be used in federally subsidized projects. Also, some states began requiring only certification while other states set minimum performance criteria if tax benefits were to realized by a consumer. For example, Oregon requires solar water heaters to save the consumer at least 50 per cent of the energy needed to supply the hot water load if it is to qualify for the state's tax credit [1]. Therefore, new standard test and rating procedures were developed which brought ICS systems and thermosiphon systems into fair competition with systems using conventional flat-plate collectors. In developing the standard to include ICS systems, it was found that it is much more difficult to rate and meaningfully compare ICS systems as opposed to active systems. In addition to the nighttime loss concern, it is not possible to measure the internal flow distribution or the internal temperature distribution. The time response of ICS systems is exceedingly long. Also, although the outdoor test environment is, of course, fully credible, it is not used because the lack of repeatability and consistency in measured performance poses a problem for systems tested at different times and locations.

The test standard which was adopted for solar hot water systems is ASHRAE Standard 95-1981 [2]. This Standard specifies the method of testing but does not specify the test conditions to be used for obtaining a standard rating. (These conditions were developed later by rating organizations.) The purpose of ASHRAE 95 is to yield results which can be used in at least three ways. First, it serves as a basis on which to compare the thermal performance of different solar domestic hot water (SDHW) systems. Second, it provides a source of information for sizing and designing solar energy systems. Finally, it can be used by a manufacturer as a development tool. The test procedure characterizes the performance of the entire SDHW system. This criterion means that the system is tested as a package with all the components installed. Collectors must first be tested under ASHRAE Standard 93-1977 [3] if the "nonirradiated array" test procedure of Standard 95 is utilized. ICS systems, which can not be tested in accordance with Standard 93, must be tested under a solar irradiance simulator as specified by Standard 95. These simulators must meet rigorous specifications. However, they are expensive, not widely available, and they limit the maximum collector area that can be tested. Also, it has been found that the thermal efficiencies of flat-plate collectors were generally higher when tested under a simulator than they were when tested outdoors because of the excessive infrared radiation exchange [4]. It is therefore necessary that tolerances on the uniformity, spectral distribution, and sky temperature be maintained and monitored to insure consistent and realistic test conditions.

For collectors that have storage outdoors, a test to ascertain the heat loss coefficient, U_L, under nonirradiated (nighttime) conditions is performed as specified by an associated rating standard [5,6]. Since this test directly follows the final day of testing for rating purposes, and the solar simulator will have just been turned off, questions arise as to the effect that the simulator might have on the unit for this heat loss test. There has been some discussion about the complexity of the current standard to rate SDHW systems. Claims have been made that the results might be too confusing for the engineer/designer as well as for the consumer [7,8], and attempts have been made to provide a comprehensive explanation of the results [9,10,11].

Tests were performed in this investigation specifically to examine the heat loss results within an indoor environment which simulates wind, sky temperature, and ambient temperature conditions outdoors. The parameters are based on actual data taken outdoors and the credibility of this comparison is examined.

1.1 SCOPE

The objective of the current research is to evaluate the significance of various parameters on the nighttime losses of ICS systems and to determine whether the usual indoor test procedure used for rating these systems satisfactorily duplicates the expected nighttime losses outside. In this investigation only one type of ICS unit was used. Several nights of outdoor data were collected on the VPI & SU campus under prevailing outdoor conditions so that the wind speed, sky temperature, and ambient air temperature effects could be studied.

Certain nights of outdoor data were simulated under the artificial environment of an indoor facility to determine how closely the heat loss could be duplicated. These tests also served to determine the accuracy of this aspect of current rating procedures.

Various parameters at the indoor facility were studied to determine which had the most effect on heat loss under an artificial environment. Tests were also run indoors which were similar to the heat loss test required by the Solar Rating and Certification Corporation (SRCC) and the Air-Conditioning and Refrigeration Institute (ARI) in their associated rating procedures [5,6] to evaluate the effects of various environmental parameters over a sixteen-hour period.

A mathematical model, based on the one developed by Lindsay [12] to model the particular ICS system used in these experiments, was then used to compare actual nocturnal cooling with a computer-aided simulation.

1.2 DESCRIPTION OF ICS SYSTEMS

The new popularity of ICS systems makes them an important class of SDHW heaters. Their simple design has been used for years by do-it-yourselfers and now many companies, including those that have primarily designed active systems, have entered the ICS market. In California an estimated 38 per cent of the 75,000 residential systems sold in 1982 were passive designs. Private industry analysts estimate that their market penetration is now approaching 50 per cent [13].

The general design of ICS systems consists of one or more relatively large black tanks located within an insulated box which has translucent glazing on one or more sides. Two classes of ICS systems which exist are concentrating and nonconcentrating types of collectors. The absorber area of the storage tank directly fills the entire glazed aperture in nonconcentrating collectors. In concentrating collectors, optical devices, such as mirrors or reflective materials, receive the unconcentrated solar irradiance and redirect it to the absorber area. The collector used in this investigation is a concentration type and it is shown in Figure 1.

Although there are many different variations in ICS design, they usually share several common features. First, the ICS units are primarily used as preheaters for conventional hot water systems. Second, the systems are generally passive; they do not require pumps, controls, or tracking mechanisms to provide hot water. Consequently, all circulation is by thermosiphon action. Third, the units are plumbed directly to city main water and the water in the tank is displaced, or drawn off, on demand. Since thermal stratification exists in the tank, the hottest water is drawn off into the residence.

The most significant difference between this type of system and an active system is that outdoor storage makes them much more dependent on weather conditions. They are a one-pass system with no recirculation of the heated fluid and the collector is exposed on cloudy days as well as at night. Their relative simplicity makes them much easier to sell and install because they come as a packaged unit and they require much less maintenance than active systems. The units can be installed on the roof or on the ground and they can be retrofit into an existing structure or built directly into new construction. Also, the cost tends to be much lower for the initial purchase, installation, and maintenance.

1.3 LITERATURE REVIEW

With the increased commercialization of solar domestic hot water heaters, standards, ratings, and warranties have become increasingly im-

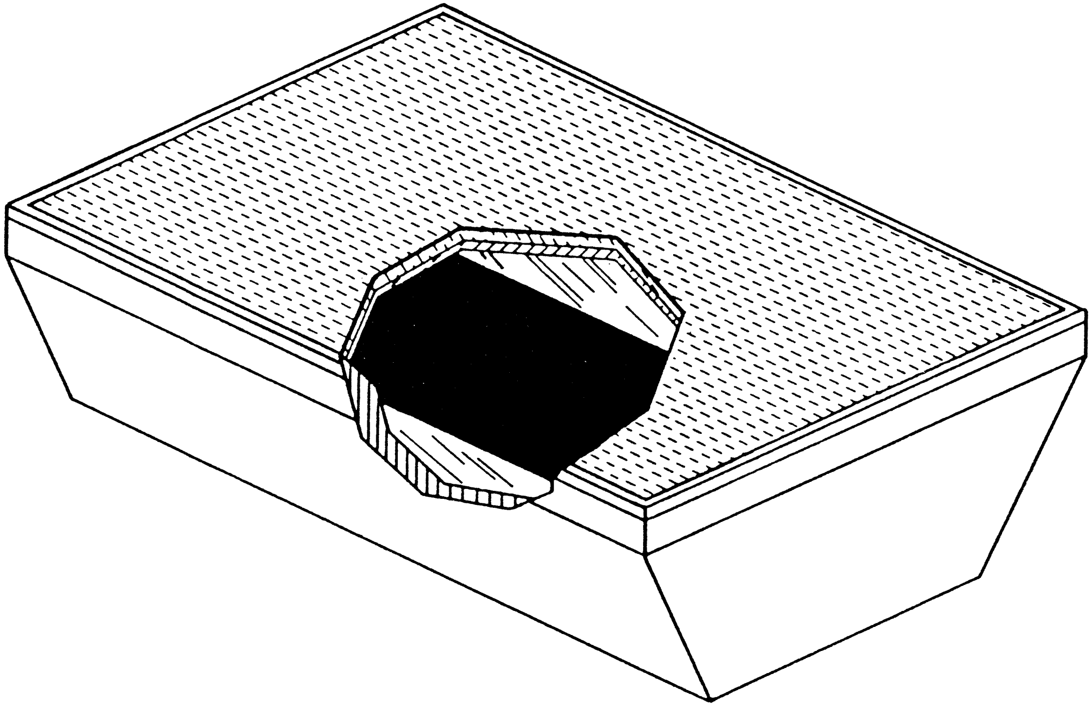

Figure 1. Cutaway View of a Cornell Energy Model 360 Concentrating ICS System

portant for the consumer, the manufacturer, and the salesperson. The most widely accepted test standard for this class of collectors is the ASHRAE Standard 95-1981 [2]. The test procedures are described by the Standard but the test conditions, including solar irradiance levels, ambient temperature profiles, and wind speed, are prescribed by an associated rating organization. Standard rating tests and procedures that have been developed by these organizations include SRCC Standard 200-82 [6], ARI Standard 920-83 [5], and TVA and FSEC standards as detailed by Goswami [9].

The ASHRAE 95 test involves three to five days of testing using a solar irradiance simulator, under controlled environmental conditions, until the daily system solar fraction has reached steady state. An important test for ICS systems, which is not described in the ASHRAE procedure, is a stagnant heat loss test. Both the ARI Standard and the SRCC Standard require such a test for all ICS systems. During this test, the collector is filled with 60 C water and is allowed to cool for 16 hours under a constant ambient temperature of 22 C and a wind speed of 3.4 m/s. A loss coefficient can then be calculated assuming a logarithmic decay of temperature with time. Since the infrared radiation environment, conventionally characterized by the sky temperature, is generally much higher inside, the credibility of these tests is questioned. Also, the results of the standards have been questioned because the systems are only tested under one set of meteorological conditions [1,14].

Some methods have been developed to try and improve the ASHRAE standard. For example, Lindsay and Thomas [15] ran experimental tests on the ICS system used in this present investigation and found that the in-line heat source approach to rating could be extended to include ICS systems, though it might not yet be feasible due to the long time constants. Methods have also been developed to extend the results from the standard tests. Klein and Fanney [14] recommended a procedure for active collectors which used the results of ASHRAE 95-1981 to determine the long term performance under different climatic conditions. Robison [1] developed a similar procedure for passive water heaters which used ASHRAE 95 results to predict thermal performance for site specific conditions based on a computer model by Reichmuth and Robison [16]. Some experimentalists thought that a short term comparative test, based on a reference system, could serve as an alternative to ASHRAE 95 testing. Cooper and Lacey [17] developed such an outdoor test method for flat-plate collectors, however, they found it to be unsatisfactory.

A number of computer simulations have been developed to predict performance for ICS systems and to optimize the design of these systems [12,16,18-23]. Cummings and Clark [18] developed a transient model for a nonconcentrating ICS system to study the thermal performance sensitivity for a variety of design options. A thorough consideration of sky temperature effects is included. Results show that a 5 - 15 per cent overprediction in performance could result if sky temperature depression effects were omitted and that nighttime insulation increased the delivered energy of a base case by about 20 per cent in cold and warm climates. Pancio and Clark [19] developed a design method based on Cummings' model and suggested the use of ambient temperature instead of an effective sky temperature, as the latter produced little change in the accuracy of their model. Work carried out at the Oregon Department of Energy (ODOE) [16,20] proposed a combination of heat up and cool down tests which would yield input parameters to a simple thermal network model that predicted performance. No account is taken for sky temperature when modeling the system but the calculated loss coefficient should be fairly accurate since it is based on data collected outdoors. Askew [21] used a lumped-node network to solve a transient analysis of a specific collector which utilized nighttime insulation and it included both the effects of convection and radiation. His model predicted the higher temperatures of the daytime well, but it overpredicted the heat loss at night.

Garg and Rani [22] presented a mathematical model for design optimization of a nonconcentrating ICS system based on energy balances which included effects of radiation to the nighttime sky. The effect of adding an insulated baffle plate inside the tank, to reduce night losses, was found to improve performance during the day as well as at night. The addition of nighttime insulation was found to increase the amount of delivered thermal energy by 70 per cent. Bar-Cohen [24] used an analytical model to optimize volume/area ratios and he stressed the need for explicit evaluation of convective and radiative losses in the prediction of nocturnal cooling. However, his study focused on the diurnal heating characteristics of of a nonconcentrating collector.

Zollner et al. [23] presented a design method which predicted long term performance, for most any ICS system, based on experimentally determined parameters. A computer model was also developed for an ICS component which is compatible with TRNSYS. An effective sink temperature was used instead of ambient temperature when determining the loss coefficient, U_L, for cases when the results were obtained indoors. There are some uncertainties associated with U_L, but the predictions appear to be good although comparisons with more experimental data would be useful.

Lindsay [12] developed the detailed mathematical model of the ICS system used in this investigation. He had good success in predicting tank temperatures during the day. His model is not easily adapted to other geometries but it has provided the groundwork for the mathematical modeling in this investigation.

Work at the New Mexico Solar Energy Association (NMSEA) [25,26,27] has provided experimental data from side-by-side tests in an attempt to improve performance and optimize design which included heat retention features and nighttime insulation for ICS systems.

Tests conducted by Jenkins and Hill [28] for flat-plate collectors were very similar to tests performed indoors in this investigation. They conducted indoor tests to determine the thermal loss characteristics while examining the effects of wind speed and sky temperature. For the same wind speed, collector efficiency was reduced for lower sky temperatures. For the same sky temperature, collector efficiency decreased with increasing wind velocity. The following conclusions regarding the control of indoor conditions, which gave the best results when compared to outdoor conditions, were reached: the wind velocity should be maintained above 3.7 m/s, and the sky temperature could range between ambient temperature and 16-19 K below ambient for the efficiency to be less than the total measurement and meteorological uncertainty. It should be noted that these results are for flat-plate collectors which do not suffer from nocturnal cooling because there is little or no fluid in the collector at night. Therefore, the range of sky temperatures is valid for their application but not necessarily for ICS systems.

The brief background review shows that while the ASHRAE Standard 95-1981 and associated SRCC and ARI rating standards could undoubtedly be improved, they will still be used extensively in the foreseeable future. Mathematical models, while showing good results in specific cases, are not likely to be incorporated as a rating standard because of a general lack of credibility. This limitation is partly because there are many different designs and there is no one good model which encompasses all of the necessary characteristics to accurately predict performance. Also, rating standards have traditionally been based on direct measurements. Therefore, immediate research is needed to reconcile questions regarding indoor testing, and specifically the credibility of the heat loss tests for ICS systems.

In Chapter 2 the experimental apparatus and procedures that were used during this investigation are presented.

2.0 EXPERIMENTAL APPARATUS AND PROCEDURES

2.1 OBJECTIVES

There were two main objectives of the experiments performed during this investigation. The first objective consisted of collecting several nights of data, under prevailing environmental conditions, to characterize the actual nocturnal performance of the ICS system. The second objective was to conduct experiments indoors, under the simulated nighttime sky environment, to duplicate system performance measured outdoors. The investigations included studying the various environmental parameters to determine which had the most significant effect on the nighttime losses of a typical ICS system. These parameters included initial fluid temperature, wind speed, ambient air temperature, and incident infrared radiation (sky temperature). A matrix of tests were also performed to determine the significance of circulating the fluid vs not circulating the fluid (i.e., letting it thermally stratify) to see how the nighttime losses were affected. If the forced circulation case did not affect the heat loss, and if it compared well with the stratified case, then either approach could be considered for use in testing procedures.

In order to fulfill the experimental objectives of this investigation, an accurate method for measuring the heat losses from the ICS system had to be devised. One method considered was to run steady-state type tests with a controlled heat source to maintain a constant temperature while monitoring the power input. The fluid would be circulated continuously in this type of test. However, this method was not feasible because of the exceedingly long thermal response of the system. (The time constant for this collector was found to be approximately 40 minutes [29].) The method which was employed to study the heat lost from the collector involved measuring the initial and final energy content of the system. However, it had to be determined whether or not these two values could accurately be obtained by monitoring only the fluid temperature in the tank at two locations. There were uncertainties associated with the temperature profile in the tank, and also uncertainties associated with the amount of energy that could have been stored in the metallic and other solid materials of tank. Therefore, a system calibration was performed in order to determine the magnitude of energy stored in nonfluid elements of the collector. If the effective heat capacity of the nonfluid elements was found to be significant, then the two temperatures could not necessarily be used as a determination of the energy content and a calibration curve would have been needed. The experimental procedure adopted to study these effects was based on calibrating the energy stored in the system vs the mixed average tank fluid temperature and is detailed in Section 2.4.1.

Several constraints and criteria were established in designing the experimental apparatus and test procedures. First, the test procedures had to apply universally to the family of ICS collectors and they also had to apply to both the indoor and outdoor systems used in this particular investigation. Second, a limitation, which applied to both the indoor and outdoor facility, was that the collector unit itself could not be disassembled to install temperature measuring devices or other instrumentation. This is because a standardized test should apply to the generic ICS system without modification. It was decided that the procedures and techniques developed in this investigation should not involve altering the internals of the collector in any manner. Therefore, although a measurement of the actual stratification in the tank was desired, only two thermocouples could be installed through existing connections. A final constraint that was established was that the data taken should be repeatable and the measurement accuracy should at least be within ±5 per cent.

To facilitate the experimental work, two test sites were used at the VPI & SU campus: the outdoor site was modified for this purpose and the indoor site was developed solely for this purpose.

The ICS system used in both the indoor and outdoor facility was manufactured by Cornell Energy, Model Number 360, and is shown in Figure 1. The dimensions and properties of the two systems are given in Table 1.

Glazing on the collector consists of three layers. The outer layer is tempered low-iron glass. The inner two layers are fiber glass acrylic. The storage tank is a nominal 0.121 m^3 (32 gal) glass-lined steel tank with a selective nickel-foil surface. The frame is fiber glass and it is insulated with reflective polyisocyanurate foam.

A special stand was built to measure the effective volume of the collector. Although the nominal volume of the tank is known, an air pocket inevitably exists because the discharge header is not physically located at the very top of the horizontal tank. Therefore, by accounting for the air space at a slope of 45 degrees, the effective volume was found to be 0.119 m^3 (31.4 gal).

Table 1. Specifications of Cornell Energy Model 360 ICS System

Specification	
Length	1.62 m
Width	1.04 m
Aperture Area	1.55 m^2
Nominal Capacity	0.121 m^3
Effective Volume	0.119 m^3
Number of Covers	3
Cover Material	Low Iron Glass (outer) Fiberglass Acrylic (inner 2)
Insulation	Reflective Polyisocyanurate

2.2 INDOOR TEST FACILITY

All indoor tests were conducted with the ICS system at a 45 degree slope on a wooden test stand as shown in Figure 2. This angle was chosen, considering the location of the discharge pipe connection, to maximize the liquid volume by minimizing the air pocket within the integral storage tank. A 45 degree tilt angle is also the required setting for the SRCC rating standards [6]. The long axis of the collector is in the horizontal position with the inlet and discharge pipes parallel to one another in the vertical plane. The test stand is located in a large room with the collector glazing facing the inside wall to prevent exposure to daylight through exterior windows. The instrumentation and the data acquisition system are also located in this room as shown in Figure 3.

Absorber plates from several surplus flat-plate solar collectors were used to construct a U-shaped sky shield and a frame was built to mount it onto the test stand. These absorber surfaces, which are what the ICS system "sees", are coated with a flat-black paint. The purpose of the sky shield is to simulate an effective nighttime sky. Its temperature is controlled by circulating a fluid through it from a constant temperature bath. The shield is large enough so that the design radiation configuration factor between the system aperture and the shield exceeds 0.90. A clearance space between the collector glazing and the sky shield was set at 0.305 m (1 ft) to allow for forced and natural circulation over the system. The entire structure is insulated with 88.9 mm (3.5 in.) thick, R-11 fiber glass insulation. The inside of the supporting frame

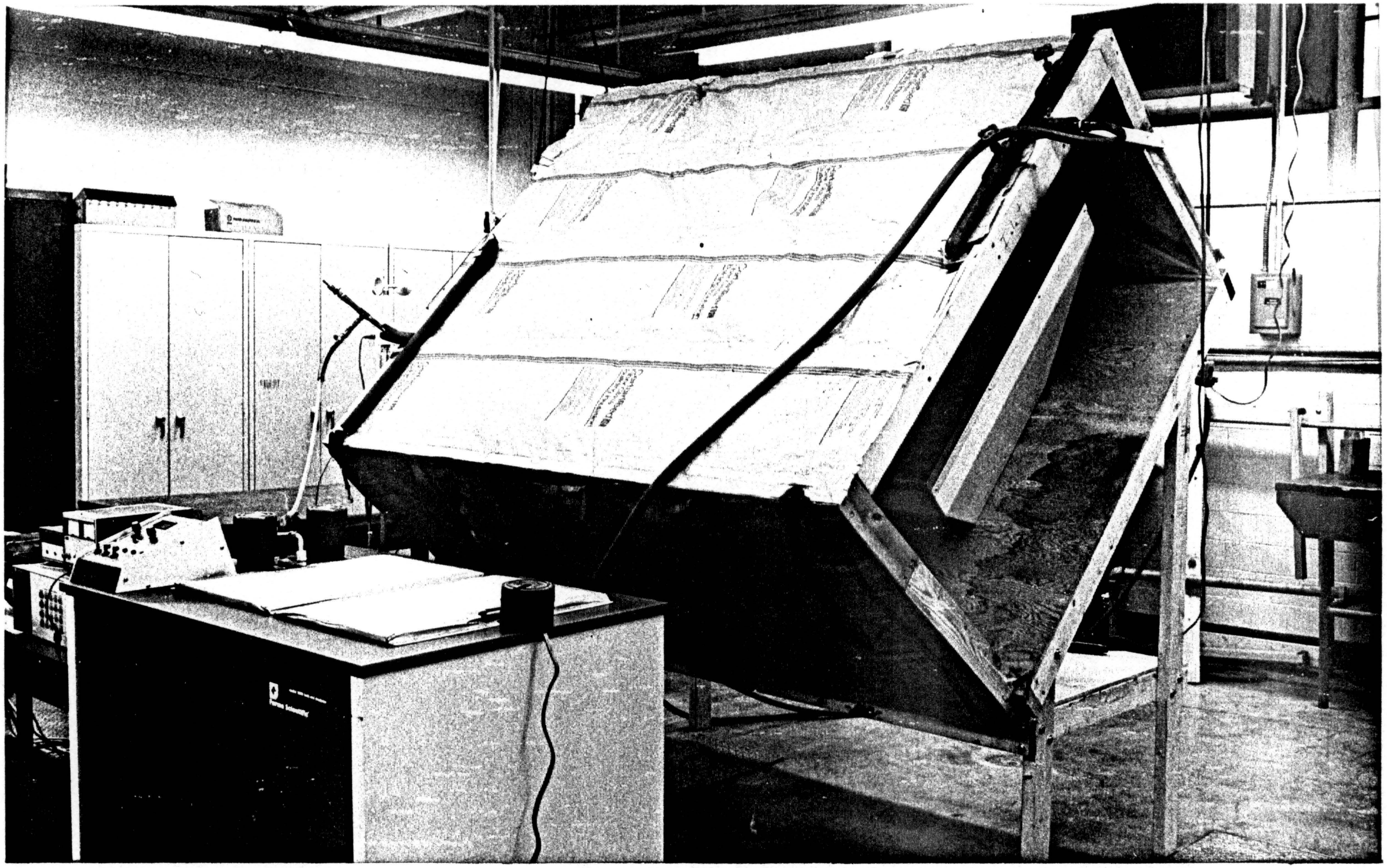

Figure 2. Indoor Wooden Test Stand with ICS System at a 45 degree Slope

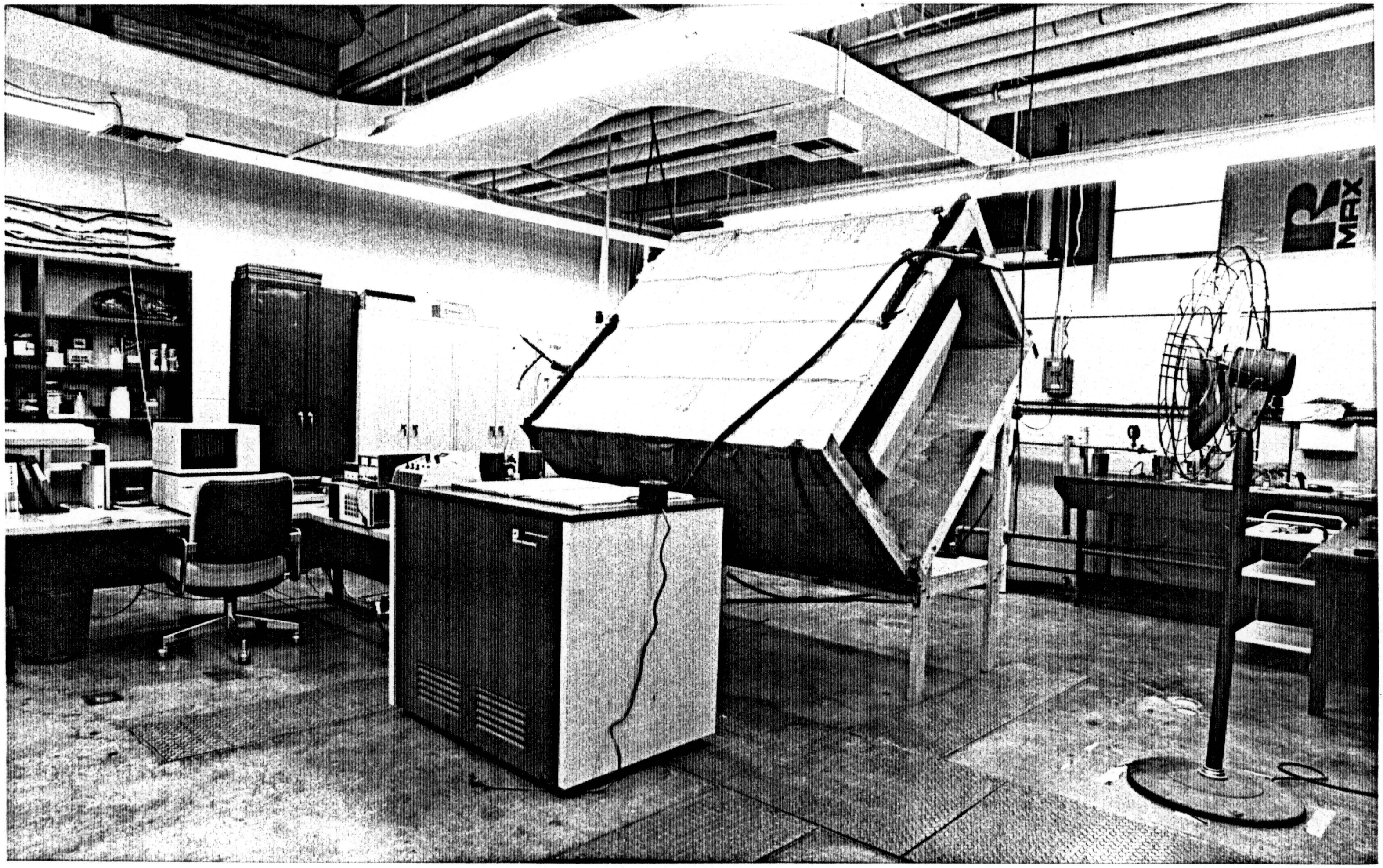

Figure 3. Indoor Test Facility

was lined with plastic to provide smooth flow when forcing air over the system.

A Forma Scientific Refrigerated and Heated Bath and Circulator provided the chilled or heated fluid for the sky shield. The circulating fluid inlet to the sky shield was on the lower side with 12.7 mm (0.5 in.) Tygon tubing and it returned from the upper side with a 25.4 mm (1 in.) hose. The fluid was a 40 per cent ethylene glycol-water solution. The volume capacity of the bath is approximately 0.189 m^3 (50 gal) and the temperature range available is -29 C to 70 C.

A listing of the instrumentation used in this investigation, along with the calibration procedures carried out, are detailed in Appendix A. The ICS system was equipped with two copper-constantan (type T) thermocouples. One thermocouple was located in the horizontal inlet header to the storage tank and the other was located in the horizontal discharge header. The thermocouple was housed in a metal sheath and was electrically insulated with shrink tubing and heat transfer compound. These sensors extended to approximately one third of the axial length of the tank and were located in a vertical plane approximately 50.8 mm (2 in.) from the tank wall.

Other instrumentation measured effective sky temperature, wind speed, and ambient air temperature. A large upright variable speed fan was used to simulate wind and was located 1.22 m (4 ft) from the test stand. Wind speed was measured 0.81 m (32 in.) from the opposite side of the test stand and at a distance of 1.68 m (66 in.) from the floor with a Weathermeasure Weathertronics Model 2011 3-cup anemometer. An Eppley model PIR pyrgeometer was used to measure infrared sky radiation. It was located on a mounting bracket at the center of the collector aperture and is shown in Figure 4 along with the insulated fluid circulating loop and some of the instrumentation. Ambient air temperature was measured with a shielded type T thermocouple located 0.99 m (39 in.) from the floor and underneath the test stand. A thermostat located in the room controlled the ambient temperature to within ±1.0 C with an air conditioning unit. Three type T thermocouples mounted at various positions on the sky shield were also monitored.

A schematic of the piping layout is shown in Figure 5. The inlet to the collector was located on the bottom side and the discharge on the top side. The Grundfos UPS 20-42 circulating pump had its suction side on the bottom inlet and delivered to the upper side. This arrangement promoted good mixing and also prevented cavitation of the pump should an air pocket have existed which exposed the upper horizontal tube at the top of the tank. A valve on either side of the pump could be used to cut it off from the loop. The pump was insulated with 19.05 mm (0.75 in.) of Armaflex insulation, and although it was a three speed pump, only the medium setting was used. The output power rating of the pump was 40 W. The piping consisted of 19.05 mm (0.75 in.) and 25.4 mm (1 in.) copper tubing and was insulated with 25.4 mm (1 in.) of fiber glass pipe insulation. A loss coefficient from the pipe and insulation, referenced to the collector aperture area, was calculated to be approximately 0.15 W/m^2 K. A 1000 W in-line cartridge heater was used to bring the system up to temperature. A pressure gauge was installed and a pressure-temperature relief valve was set at 0.689 MPa (100 psi).

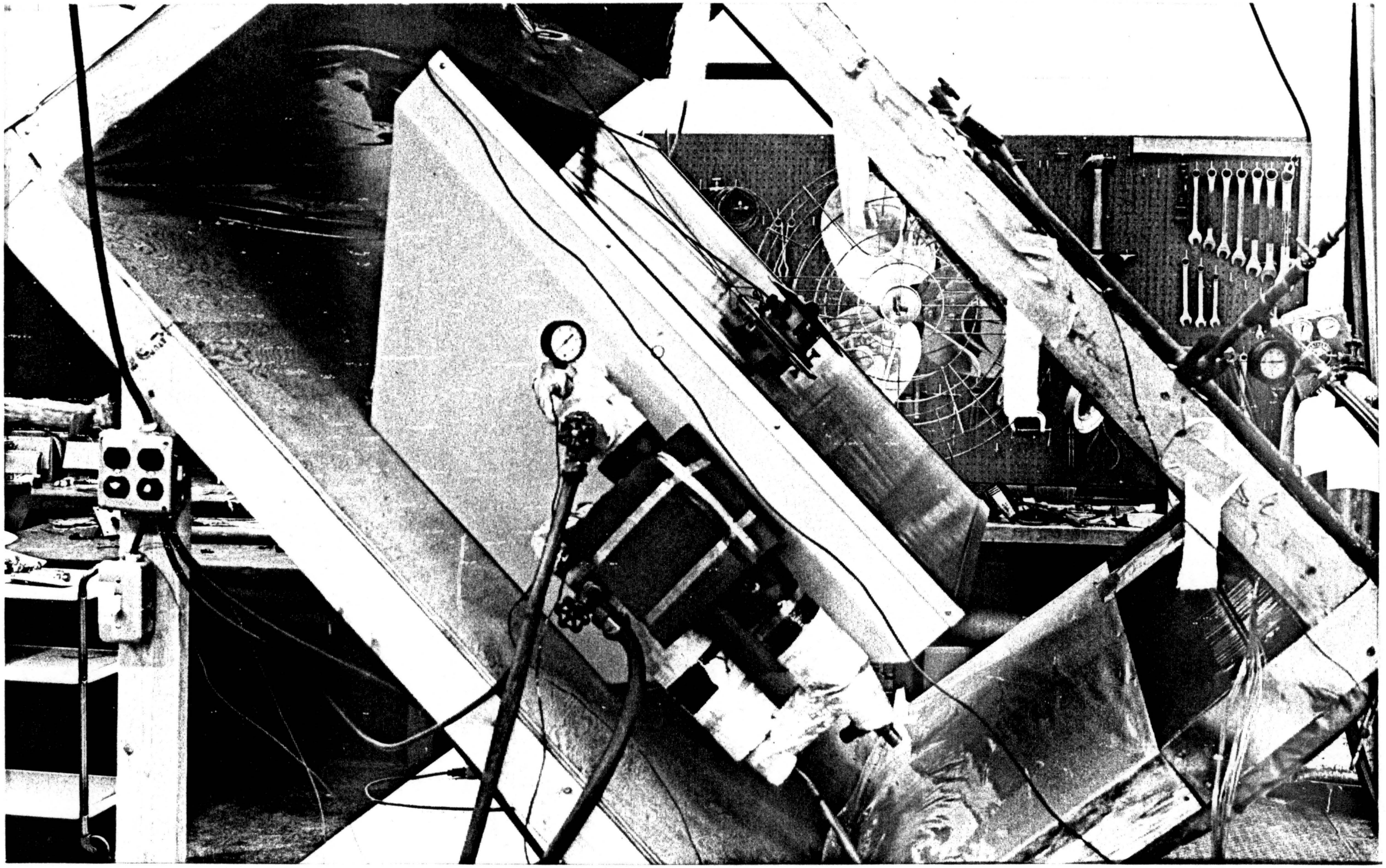

Figure 4. Side View of the Indoor ICS System and Instrumentation

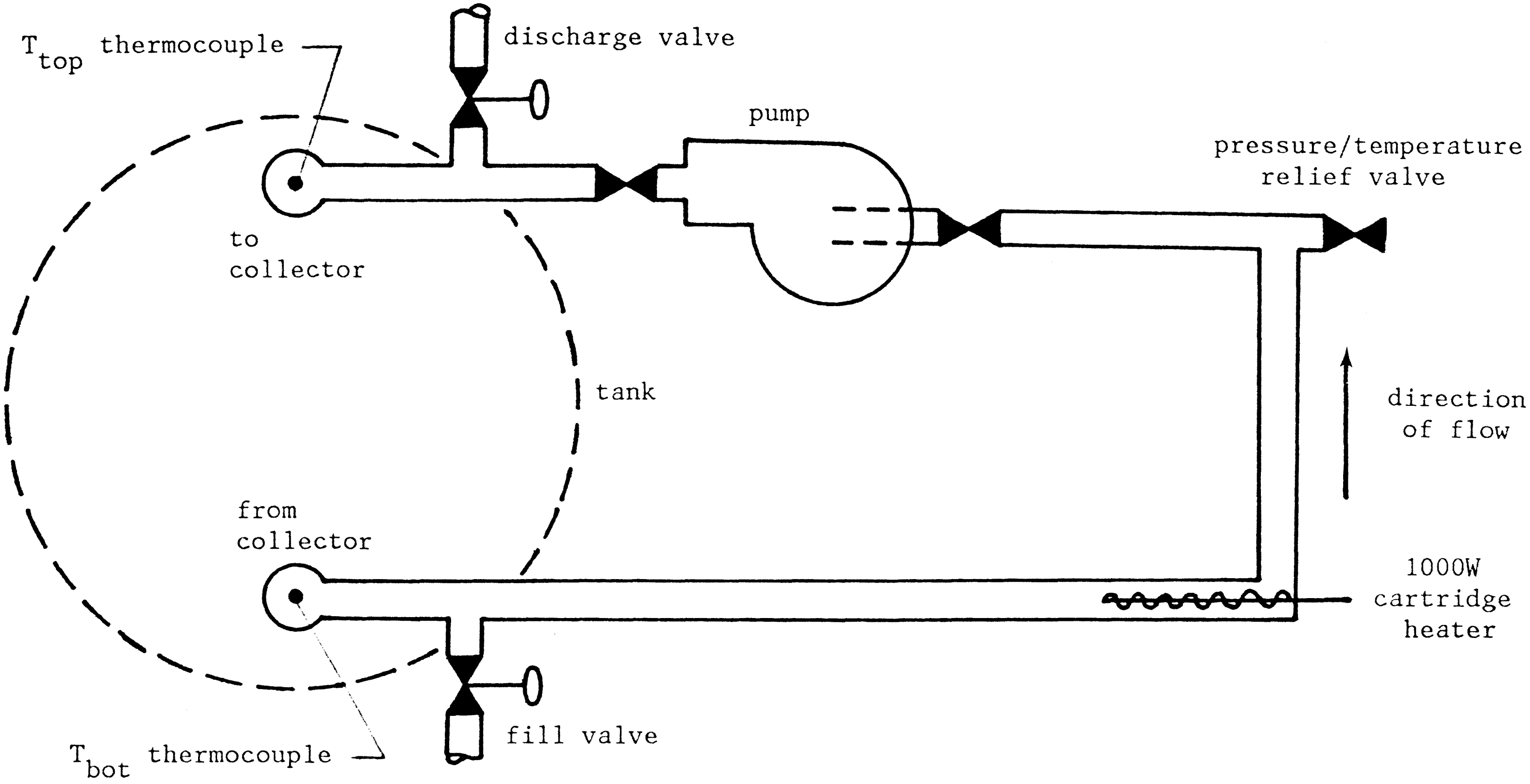

Figure 5. Piping Schematic of Indoor Test Collector

The results were monitored and recorded with a Hewlett Packard HP3497A Data Acquisition/Control Unit. The thermocouples from the storage tank, ambient air, and sky shield were connected to a terminal card which was configured for hardware compensation of type T thermocouples. The anemometer and the pyrgeometer were connected into a terminal card of an analog signal multiplexer which read low-level DC voltage. At 5-minute intervals, the channels were scanned and the data was read into a Hewlett Packard HP9236 microcomputer. The computer program which activated the HP3497A then converted these analog voltage signals into physical quantities. The results were displayed on the CRT, recorded on a parallel printer, and stored on a flexible disk. After completing an experiment, the stored quantities were transferred to the VPI & SU IBM 3084 mainframe computer for reduction and permanent storage. All data collection and transfer were performed with the aid of computer software and therefore eliminated the opportunity for human error in measuring and entering these quantities.

2.3 OUTDOOR TEST FACILITY

The ICS collector used at the outdoor facility was also a Cornell Energy Model 360 unit, as shown in Figure 1, with the same design characteristics as the indoor collector.

The ICS system was mounted on an adjustable tilt test stand, at an angle of 45 degrees, as shown in a front view in Figure 6. The collector was located at a latitude of 37.23 degrees north, on the roof of a large building (Randolph Hall), and faced due south. The cables from the instrumentation ran down to a data acquisition system located within the building several flights below.

The outdoor test facility was equipped with instrumentation to record infrared sky radiation, wind speed, wind direction, ambient air temperature, rainfall, and temperatures within the storage tank. A list of the instrumentation, giving the manufacturer, specifications, and calibration results are included in Appendix A. The instrumentation room for the outdoor facility is shown in Figure 7.

Again, the type T thermocouples used in the storage tank were only located in the inlet and discharge connections of the collector. They were inserted to approximately one-third of the axial length of the tank, and were located in a vertical plane approximately 50.8 mm (2 in.) from the tank wall. Wind speed was measured with a three-cup model W103B Weathermeasure anemometer. The effective sky temperature was measured normal to the aperture plane of the collector with an Eppley model PIR pyrgeometer. The ambient air temperature was measured with a type T thermocouple located in a white, ventilated box. A Weathermeasure lightweight vane, model W104-2, was used to measure wind direction. A calibrated rain gauge located near the test stand was used to measure nightly rainfall. A side view of the outdoor test facility is shown in Figure 8.

The schematic of the piping configuration is exactly the same as the indoor configuration, shown in Figure 5, with the exception of the pressure gage which was not used. The heater used outdoors was the same as the indoor heater but was controlled with a Variac variable AC autotransformer. The Variac was used for convenience because most of the

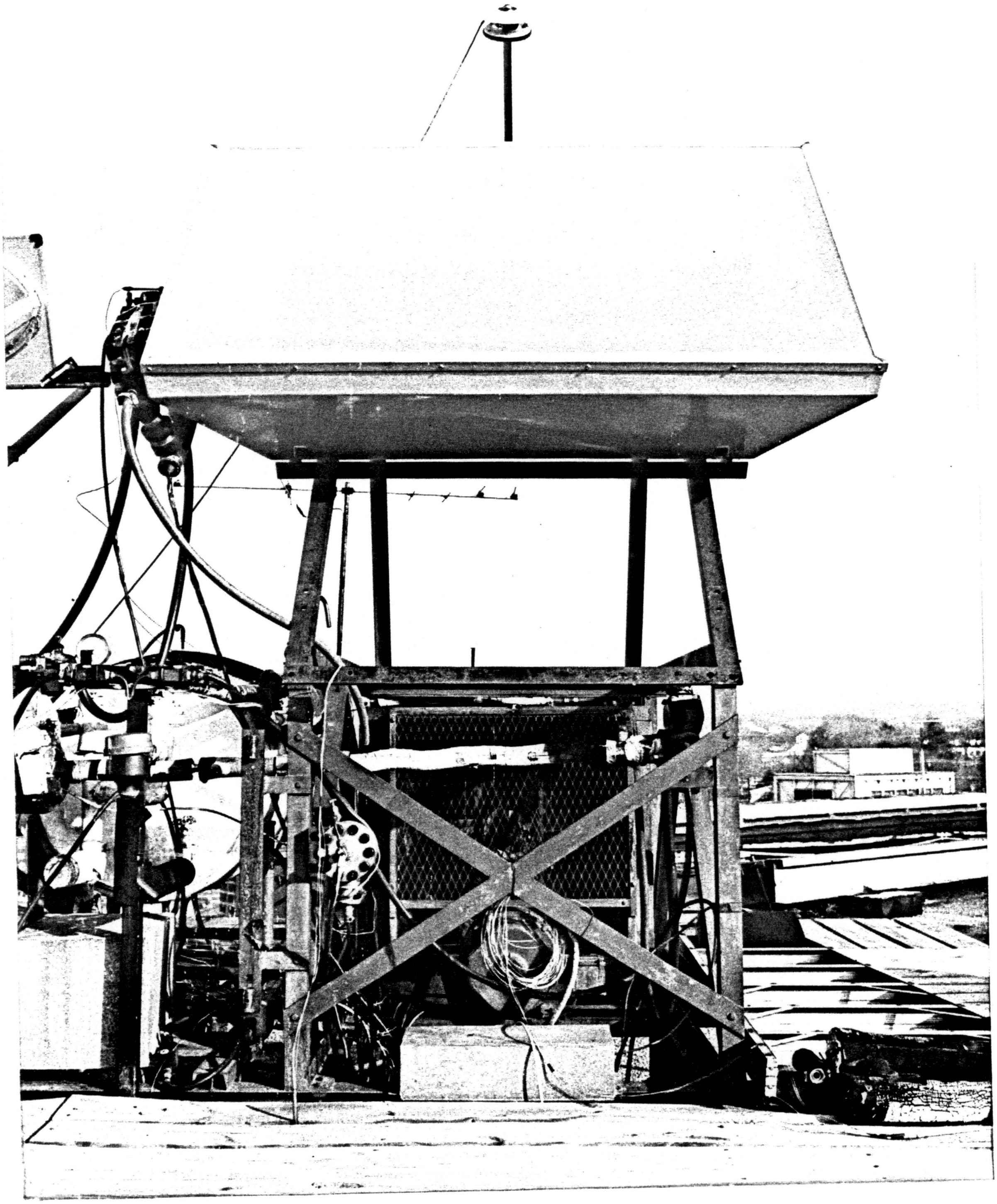

Figure 6. Front View of Outdoor ICS System

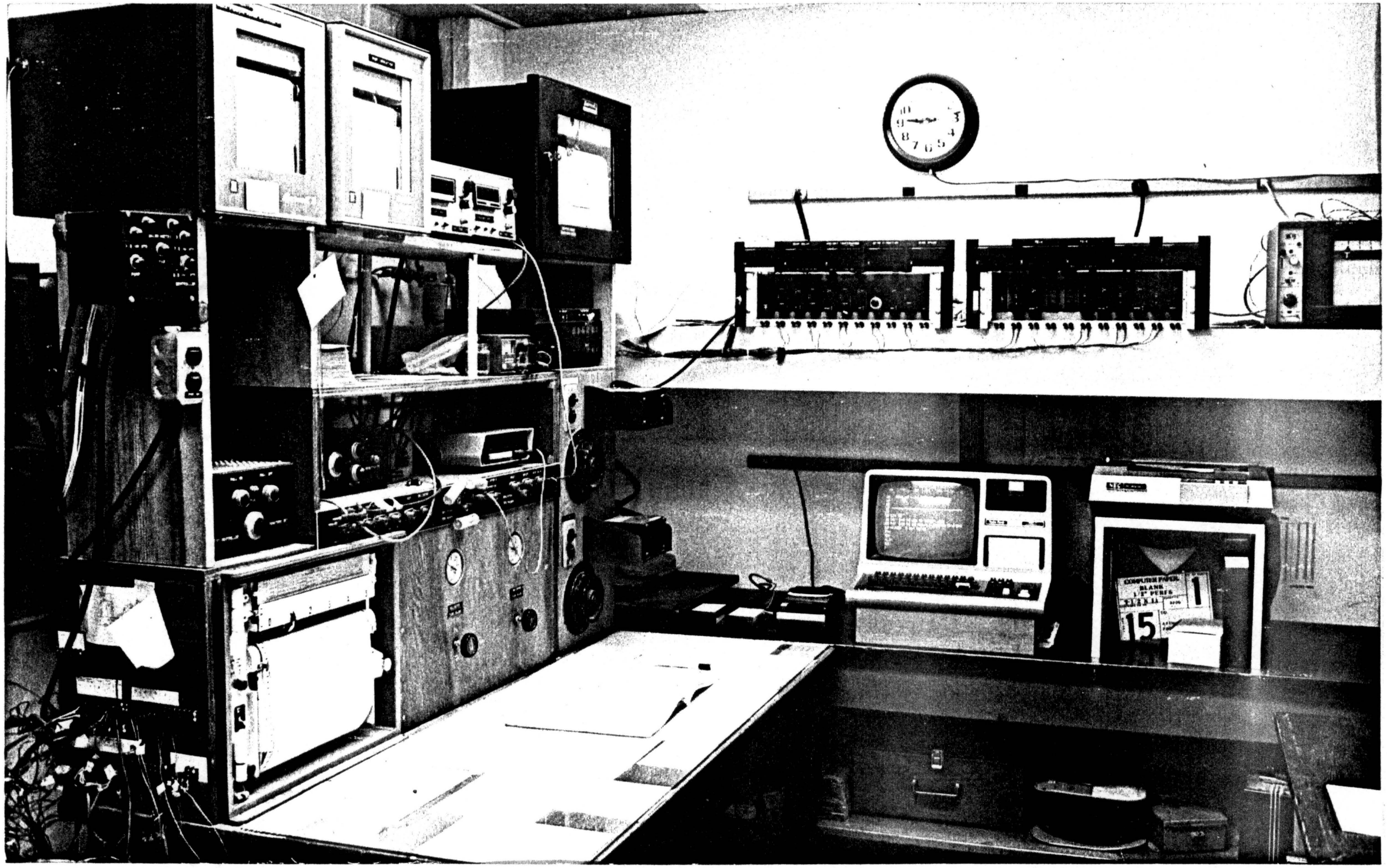

Figure 7. Instrumentation Room for Outdoor Test Facility

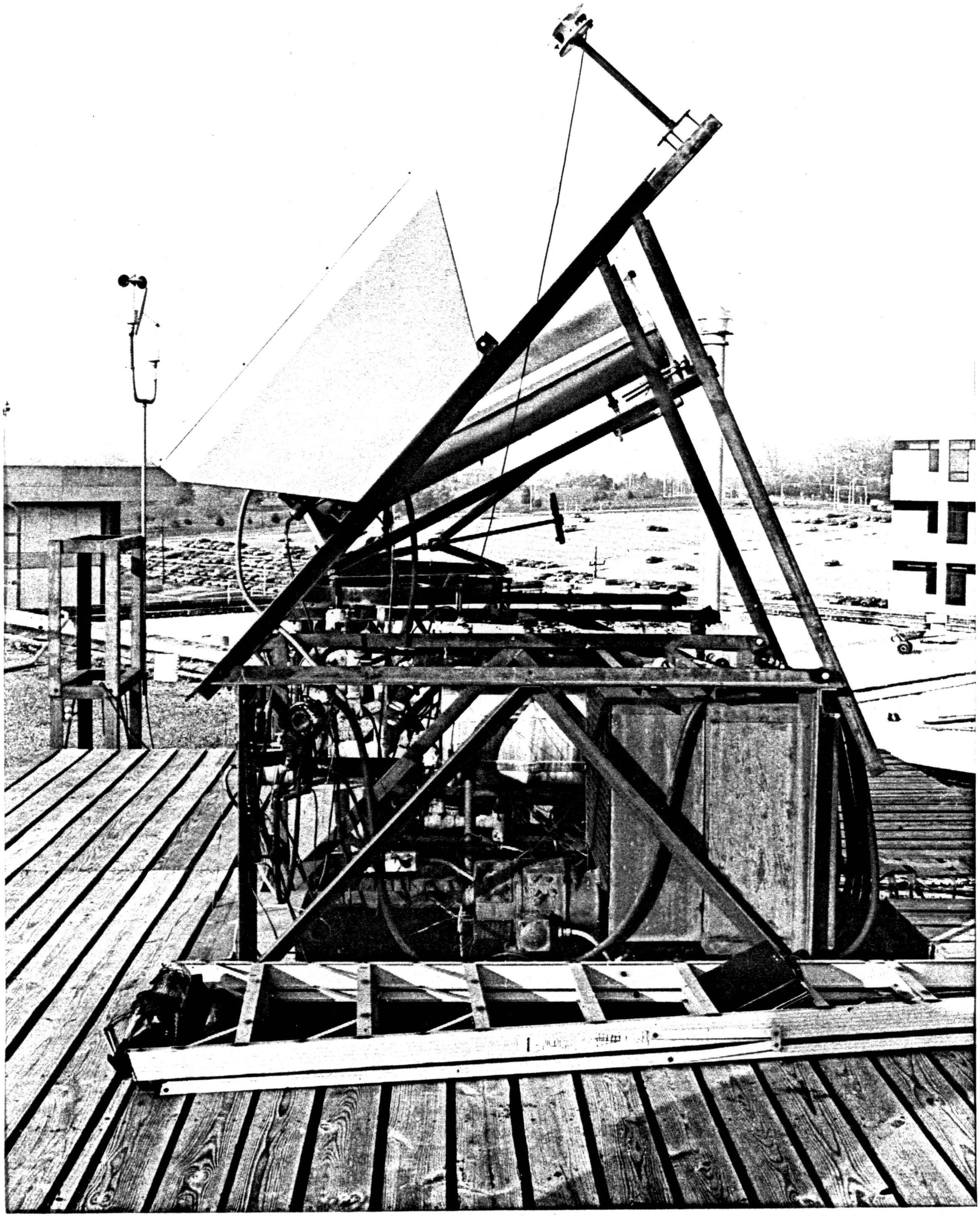

Figure 8. Side View of ICS System and Outdoor Test Facility

energy was supplied naturally by the sun and only small amounts of heat addition were required to obtain the desired initial temperatures. The piping was insulated with 19.05 mm (0.75 in.) of Armaflex insulation.

The thermocouples from the storage tank and the ambient air as well as the cables from the anemometer, weathervane, and pyrgeometer were all connected to an automatic data acquisition system. Certain of these quantities can change drastically within a few seconds due to the unpredictable nature of an outdoor environment. Therefore, these signals were sent to electronic integrators, manufactured by AGM Incorporated, where they were integrated over a 5-minute period. The integrators were then reset on 5-minute intervals and this average signal was then recorded by a Radio Shack TRS-80 Model III microcomputer. The computer program which activated this reset then converted the voltage data into physical quantities. The results were displayed on the CRT and a hard copy was generated by an attached printer. A cassette recorder was also connected to the TRS-80 and, at the completion of a test, the data was recorded onto the cassette. These stored quantities were then transferred to the VPI & SU IBM 3084 mainframe computer for reduction and permanent storage. Again, all data was collected and transferred without human intervention.

2.4 EXPERIMENTAL PROCEDURES

2.4.1 SYSTEM CALIBRATION

A system calibration was performed in order to determine the magnitude of energy stored in the nonfluid elements of the collector. The indoor

ICS system was used for this calibration and it was mounted on the indoor test stand at a 45 degree angle as described earlier. Thermocouples were located in the horizontal inlet and discharge connections to the collector.

An "adiabatic" box was utilized in this calibration procedure. The reason for using this type of box was that the energy of the displaced water in the box could be assumed to be essentially the energy stored in the fluid and nonfluid elements of the collector. This is because the effective thermal mass was very low due to its construction, and losses from the box for the time period of the test could be considered negligible. The box had an insulated and lined lid to reduce evaporation and heat losses from the top, and a small hole was cut into it to allow insertion of the discharge hose from the collector. The hose fed in the water at the top of the box to promote good mixing and prevent stratification.

The dimensions of the outside of the box were 0.686 m x 0.686 m x 1.219 m (27 in. x 27 in. x 48 in.) and it consisted of a 12.7 mm (0.5 in.) thick plywood frame. Rigid polystyrene insulation of 50.8 mm (2 in.) thickness was located along the insides and the inside bottom of the box to minimize the heat capacity. The volume of the inside of the box was 0.361 m^3 (95.3 gal). A 0.10 mm (4 mil) thick polyethylene liner was used to form a leakproof barrier, of negligible heat capacity, that prevented water from seeping into the insulation. A thermocouple tree was installed in the center of the box which held three equally spaced thermocouples. An average of these three readings was taken to determine the final temperature of the displaced water.

The system calibration test began by initially filling the ICS tank with hot water and then heating it further to a specified temperature. The adiabatic box was placed on a set of scales and the initial dry mass was measured. The pump was kept on to circulate the water through the collector tank loop and to allow the temperatures to stabilize. Once equilibrium was reached a computer program was started to record data at 10 second intervals and the circulating pump was turned off. Since the fluid was well mixed, and had a nearly uniform temperature throughout, the initial temperature was determined by taking the algebraic average of the two fluid temperatures within the tank. Immediately cold tap water, at a constant flowrate and temperature, was added into the inlet of the collector and displaced the fluid in the storage tank into the nearly adiabatic box. This draw continued until the temperature at the outlet was within ±1 C of that for the inlet. At the end of the displacement, the final mass was read and the difference between the initial dry mass and final mass gave the overall mass of water displaced. The mass of the displaced water and its average mixed temperature were recorded, and used to calculate the total energy of the system. This quantity was then compared to the energy initially stored in the fluid element only. The ratio was calculated and the entire procedure was repeated for several different initial temperatures. In each case it was found that the ratio was within ±4 per cent of unity. This value is probably within the limit of accuracy of the measurement technique. It was concluded therefore that the energy stored in the collector could be accurately calculated, provided that the fluid was well mixed, by taking the algebraic average of the two tank thermocouples which were installed in the inlet and discharge connections. When the fluid is well mixed the two fluid temperatures are within a fraction of each other. Therefore, in this case, either temperature could be used, but taking the algebraic average increases the accuracy slightly.

2.4.2 TEST PROCEDURES

The realization of the two objectives of this experimental investigation required two types of tests; indoor tests and outdoor tests. The indoor tests were conducted at any time of the day or night. There were three categories of indoor tests: stratified tests, forced circulation tests, and SRCC tests. The outdoor tests were scheduled so that it would be dark during the entire test period. This criterion was to prevent any shifts in the environmental parameters that might be due to the presence of the sun. It was desired that all types of environmental conditions be examined outdoors. However, little data was taken on rainy nights as this is not an easy phenomena to analyze in terms of heat loss. Two categories of outdoor tests were run: stratified tests and forced circulation tests.

The outdoor test preparation began with servicing the ice bath for the thermocouple reference junction and adjusting the initial temperature of the tank either through heating or the addition of cold water. Once this was set, the circulation pump was turned on until the contents of the fluid in the storage tank were fully mixed and at a nearly uniform temperature. The purpose of mixing the tank contents right up to the first reading was so that the temperature of the water would be nearly uniform and the initial temperature corresponded to the algebraic average of the two tank thermocouple readings.

The test commenced when it was dark outside. The computer program was started and, for the stratified test, at the first 5-minute interval the pump was turned off. The collector was left alone to dissipate heat for eight hours, under the prevailing environmental conditions, while the two storage tank temperatures, ambient temperature, sky temperature, wind speed, and wind direction were recorded by the computer every 5-minutes. After the 96th reading (8 hours), the pump was turned on manually to thoroughly mix the contents of the tank for 4 more readings. After the final mixing was concluded, the test ended.

At the end of a stratified test (the 96th reading), the two tank thermocouple readings were separated by a couple of degrees because, due to thermosiphon action, the contents had stratified. It was desired to obtain an accurate value of the final fluid temperature for heat loss calculations so the contents were mixed to see how the mixed temperatures compared with the stratified temperatures. At first one might assume, for the stratified case, that the temperature profile would be linear between the top and bottom thermocouple readings and that the final temperature would be the algebraic average of these two quantities. However, after examining the mixed fluid temperature of reading numbers 97 - 101, it was found to not be so. In fact, the final average temperature was weighted heavier towards the bottom thermocouple reading. Several cases were examined and the results were found to be the same. This finding was also confirmed by Lindsay [12], who performed tests on the same type of system but with the addition of three more thermocouples inside the tank. It was concluded therefore that a weighted average of 1:2,

$$T_{w,avg} = \frac{T_{top} + 2\ T_{bot}}{3} \tag{1}$$

for the 96th reading, could be calculated to determine the final temperature. This situation holds true for both the indoor and outdoor facilities.

The other category of outdoor tests that were performed is the forced circulation test. In this test, the same procedure is followed for the stratified test except that the pump is not turned off and it is left to circulate continuously for the entire eight hour period. The final 4 cycles of mixing were performed, although not necessary, and the exact same parameters were monitored.

The indoor test preparation began with the system filled with water. The water was either heated by adding energy with the in-line heater or cooled with cold water addition until the desired temperature was reached. The windspeed, the chiller temperature (which provided the flow for the sky shield), and ambient air temperature were set and allowed to come to equilibrium. The tank water was continually mixed until the test began. When all environmental parameters were at the desired levels, and had reached steady state, the computer program was started and the test commenced. For the stratified test, the pump was turned off at the first 5-minute interval. The system was left alone to dissipate heat naturally for eight hours while the two tank temperatures, wind speed, ambient temperature, sky temperature, and three sky shield temperatures were recorded every 5 minutes by the computer-controlled data acquisition system. After the 96th reading (8 hours), the pump was turned on again to thoroughly mix the contents of the tank for 4 more readings. After this mixing period was concluded, the test ended. During the first several experiments, the pump was controlled manually whereas later the pump was controlled by relays which were activated by the computer software.

Forced circulation tests were also conducted indoors. This type of test is identical to the stratified test except that the pump is not turned off after the first reading. Instead, it is left to circulate for the entire eight hour period. The final 4 cycles of mixing were performed, although not necessary, and the exact same parameters were monitored.

The indoor tests that are entitled SRCC tests in this investigation were similar to the heat loss test required by the SRCC Standard 200-82 [6]. These tests were conducted with the same ambient and initial temperatures, and run for 16 hours, but with varied wind speeds and sky temperatures. This test is identical to the stratified test except instead of leaving the collector to sit for 8 hours, it is left to sit for 16 hours.

A series of indoor and outdoor tests were run between the months of September and November so that a wide variety of environmental conditions would be encountered. Approximately 30 tests were performed outdoors and 34 tests were performed indoors. Although, not all of the the test results are included, as some of the cases were very similar, the majority of the results are examined in Chapter 4.

3.0 ANALYTICAL PROCEDURES

3.1 MATHEMATICAL MODEL

A mathematical model was developed by Lindsay [12] to simulate the solar energy collection process in an ICS system. The objective of this portion of the current investigation was to utilize this previously developed model to determine how well it could predict nighttime losses. For the diurnal heating process, there is at least a 10 C increase in temperature within the tank and the incoming solar radiation is much greater than the energy lost. The differences that might affect the accuracy of the model for studying the nighttime losses are that the change in temperature tends to be less than 10 C, there is no solar irradiance input, and the lost energy is equal to the change in stored energy. The model's accuracy is therefore put to a more stringent test.

If the model was found to compare well with the experimental data, as it did for daytime collection, it was desired to examine different environmental parameters to determine their relative contribution to the associated decrease in tank temperature. The model utilized a simplified theory with inputs of test derived parameters for the collector elements.

A brief outline of the essential features of the model are presented in this chapter. Further details relating to the calculation of the overall loss coefficient, the various temperatures, and the radiosities are presented in Appendix B. More information regarding the intricacies of the model can be found in reference 12.

3.1.1 FORMULATION

Due to the changing climatic conditions, and the thermal response time of the water in the tank, the behavior of the ICS system is transient. The development of a model to simulate this type of performance can be highly complex. Consequently, several simplifying assumptions must be made. One very important assumption that was made in this investigation was that the effective heat capacity of the ICS system was taken to be just the fluid in the tank. Specifically, the heat capacity of the cover system and the internal reflectors were neglected. This simplification was necessary in order to allow the tank to be considered as a single node for which an average hourly temperature can be determined. Also, in analyzing the absorber-tank of the ICS system, the water was assumed to be fully mixed, i.e., no thermal stratification.

An energy balance can be written about the absorber tank as

$$\begin{matrix} \text{stored} & = & \text{collected} & - & \text{delivered} & - & \text{lost} \\ \text{energy} & & \text{energy} & & \text{energy} & & \text{energy} \end{matrix}$$

The stored energy is based on m_e and C_e, the effective mass and specific heat of the tank, respectively, and dT_t/dt, the rate of change of the tank average water temperature. The collected energy is the absorbed solar radiation into the tank which is equal to zero for the study of nighttime losses. The delivered energy term is also equal to zero because in this particular investigation, no draws were performed. The lost energy, q_{loss}, to the ambient air can be represented by an overall loss coefficient, U_L, and the area of the collector aperture, A_c, defined as

$$q_{loss} = U_L A_c (T_t - T_a) \tag{2}$$

Consequently, for the study of this investigation, this energy balance reduces to

$$m_e C_e \frac{dT_t}{dt} = - U_L A_c (T_t - T_a) \tag{3}$$

Integrating equation (3) with respect to time, with an initial temperature T_0, yields

$$T_t = T_0 \exp(- \frac{U_L A_c}{m_e C_e} \zeta) + T_a [1 - \exp(- \frac{U_L A_c}{m_e C_e} \zeta)] \tag{4}$$

In this investigation, it was desired to compare the simulated results with the experimental results which were hourly average data. Therefore equation (4) was integrated over a time interval, $\Delta\zeta = \zeta_k - \zeta_{k-1}$, which was chosen to be one hour.

$$\overline{T_t} = \frac{\int_{\zeta_{k-1}}^{\zeta_k} T_t \, d\zeta}{\zeta_k - \zeta_{k-1}} \tag{5}$$

Integration results in an equation for the hourly average temperature, i.e.,

$$\overline{T_t} = T_a + \frac{(T_0 - T_a) m_e C_e}{U_L A_c \Delta\zeta} [1 - \exp(- \frac{U_L A_c}{m_e C_e} \Delta\zeta)] \tag{6}$$

where T_0 is the temperature at the beginning of each hour.

The quantities that are unknown are m_e, C_e, and U_L. The overall loss coefficient is strongly temperature dependent, as is C_e to a lesser extent. The effective mass of the water in the tank can be determined by accounting for the air space at a slope of 45 degrees from the effective volume as discussed in Section 2.1. The effective specific heat of the tank was taken to be equal to the specific heat of the fluid in the tank because the energy stored within the nonfluid elements was found to be negligible, as discussed in Section 2.4.1. The overall loss coefficient can be expressed in terms of the convective and radiative losses from the absorber-tank in an iterative solution procedure using the results of Appendix B.

3.1.2 SOLUTION TECHNIQUE

A FORTRAN computer program was employed to model the ICS system. First, hourly average values of the tank temperature, wind speed, sky temperature, and ambient temperature are read in for each of the eight hours of the test period that is being simulated. Next, the diffuse transmittance and reflectance of the covers are calculated and the radiation configuration factors are determined, as these quantities do not change throughout the program.

The remaining calculations involve an iterative procedure in order to solve for the losses from the collector. A mean tank temperature is assumed and the temperature of the collector components are estimated. This set of temperatures is then allowed to converge through a two step iterative procedure which determines the radiosity distribution and the energy balances on the collector components. Once these temperatures converge, the lost energy can be calculated and the overall loss coefficient determined from equation (2).

Further details of the calculation procedure can be found in Appendix B.

3.2 SKY TEMPERATURE ANALYSIS

In the development of the mathematical model in the previous section, the lost energy was referenced to ambient temperature and an overall loss coefficient. The losses from ICS systems are dependent on convection and radiation through the cover system and, to a much lesser extent, through the sides and back of the collector. The convective losses are to ambient air temperature and the thermal radiation losses are to an effective sky temperature. While the ambient air temperature is a well known quantity, the sky temperature is much less familiar. In the following paragraphs the sky temperature will be defined and the uncertainties associated with this parameter will be discussed. An explanation will also be given as to why the lost energy can be referenced to the ambient temperature even though the losses are through two different paths.

The sky temperature is assumed to be an equivalent blackbody temperature and the emissive power is characterized by the following equation.

$$E_s = \sigma T_s^4 \tag{7}$$

The atmosphere is partly transparent to radiation, within the infrared spectrum, in the wavelength region between 8 µm - 14 µm. The atmospheric radiation at ground level is based mainly on that from water vapor and carbon dioxide, with water vapor being the more important of the two. The sky temperature is found to be lower than the ambient temperature because the atmospheric temperature decreases with elevation. Consequently, the largest difference between the sky temperature and ambient temperature occurs on a clear night.

Although radiation to the sky has received limited attention with regard to flat-plate collectors [30,31,32,33], little attention has been given to its relationship with ICS systems [23]. However, nocturnal cooling, while presenting a problem for ICS systems, has been studied and utilized as a method of dissipating heat at night for the cooling of buildings [34,35,36]. In these applications the same system can be used to heat in the winter and cool in the summer, thus taking advantage of both the daytime heating and nighttime cooling effects.

Several relationships have been proposed [37,38,39] to relate the sky temperature to the ambient temperature and other environmental parameters. Thomas and Reed [33] present five such correlations. They found that for a given ambient temperature, the various correlations predicted sky temperatures that varied by up to about 20 C. This suggests that the sky temperature is a difficult parameter to characterize, and it is dependent on many other environmental variables.

The emitted and reflected irradiance from the ground surface strongly affects the incident infrared irradiance on a collector with a tilted aperture. This further complicates the determination of an effective sky temperature. The measurements taken outdoors with a pyrgeometer, etc. account for this effect because it "sees" what the collector "sees". However, when modeling the sky temperature the effects of the ground are not usually included and this results in uncertainties associated with its determination. In the computer model used in this investigation actual hourly values for the sky temperature, the ambient temperature, and the wind speed, taken from outdoor data, were used as input variables. Therefore, ground effects were accounted for in this work.

The development of an effective sink temperature, T_e, to be used instead of the ambient air temperature when calculating the losses as in equation (2), has been suggested by a number of sources [23,24,30,31,32]. Since the thermal radiation losses and the convection losses are to two different sink temperatures, T_s and T_a, respectively, it was desired to obtain a single parameter which would replace these two and yield more accurate results. The suggested equation for T_e includes the radiation heat transfer coefficient, the external convective heat transfer coefficient due to wind, back and edge losses, as well as T_a and T_s [31]. Some methods suggest that for a well designed collector, the back and edge losses can be eliminated. A second simplification also results when the radiative heat transfer coefficient is taken to be one-third of the convection coefficient due to wind [30], thus reducing the equation for T_e to

$$T_e = T_a - 0.25\ (T_a - T_s) \qquad (8)$$

Then this value of T_e is substituted for T_a into equation (2) to yield

$$q_{loss} = U_L A_c (T_t - T_e) \tag{9}$$

At a given time, U_L can be calculated from equation (2). In this manner, it can be seen that U_L is an arbitrarily defined overall loss coefficient based on a certain reference ΔT. If the ΔT is altered, then U_L will also be changed. Therefore, how it is defined is not as important as making sure it is being used consistently.

In cases when U_L is determined from an indoor test which does not include sky temperature effects, its value can be expected to be smaller than would result naturally outdoors. If it was desired to use this value of U_L to determine q_{loss} for an outdoor case (using equation (2)), then a smaller value of q_{loss} would result as well. However, if T_e was determined from equation (8), with outdoor values of T_a and T_s, then q_{loss} could be determined as in equation (9). The value of q_{loss} determined from equation (9) would tend to be greater than that found from equation (2) because T_e is less than T_a and a larger value of ΔT results. Therefore, using T_e with values of U_L that were obtained indoors may yield more realistic values of q_{loss} for outdoor tests. However, in the case of this investigation, since the parameters were obtained from actual outdoor data, U_L was defined in the conventional manner.

In this investigation, the effects of sky temperature on the nighttime thermal performance of ICS systems were studied. The current rating standards require loss tests with an ambient temperature of 22 C, and a wind speed of 3.4 m/s, but they do not mention any tolerance on sky temperature variation. As mentioned earlier, the sky temperature indoors tends to be higher than the ambient temperature, which is an unrealistic natural outdoor condition at night. Therefore, collector designs which include heat retention features might not even be tested in the range where their positive attribute would be realized. If these rating standards are used heavily by consumers in selecting collectors, the ICS systems tested which do not include any heat retention features may enjoy a favorable, but unfair and unrealistic, marketing advantage.

The results from the indoor and outdoor experiments and the results from the mathematical model are presented in the following chapter.

4.0 RESULTS AND DISCUSSION

An optimal way to study the heat loss from an ICS system would be to take a variety of data outdoors and to correlate all the environmental parameters into one so that all the tests could be compared on the same basis. Unfortunately, a reliable means of combining all the parameters into one "effective parameter" has yet to be developed and it is not possible to hold an environmental parameter constant outdoors so that another may be varied.

The results in this section present several sample outdoor test runs and comparisons in Figure 9 through Figure 15. The results of comparisons between indoor tests and outdoor tests are presented in Figure 16 to Figure 26. A study of parameter variation in 8 hour indoor tests is presented in Figure 27 to Figure 30. Figure 31 through Figure 34 show the results from 16 hour indoor SRCC type tests. In Figure 14 to Figure 34, the results are plotted as a function of time in terms of a nondimensional temperature variable, θ, defined as

$$\theta = \frac{T - T_a}{T_0 - T_a} \tag{10}$$

(All of the data points that are plotted in Sections 4.2 through 4.4 are for actual experimental data but they are plotted as a continuous line. The reason for this is that there is a data point every five minutes and the graphs would be more difficult to read if symbols were used.)

As mentioned earlier, heat loss tests that were performed outdoors were scheduled so that it was dark for the duration and they usually were conducted in the middle third of the night. The starting time of the test, the initial and final tank temperatures, and the average hourly data are tabulated in Appendix C for each test discussed. A summary of the overall results for each test are listed in Table 2. The values of wind speed, ambient air temperature, and sky temperature that are listed in Table 2 are all averaged quantities over the eight-hour period. For the outdoor tests these parameters can change drastically, whereas for the indoor tests they are essentially constant throughout the test period.

4.1 OUTDOOR TEST RESULTS

One of the main objectives of the experimental tests was to determine the relative effect of the sky temperature, the wind speed, and the ambient air temperature on the heat losses from the ICS system. It is difficult to arrive at accurate conclusions when comparing outdoor environmental data because these three environmental parameters can change between different nights of testing and they also can vary significantly within the same night of testing. Plots of averaged hourly quantities for outdoor heat loss tests are presented in Figure 9 through Figure 12 and they show the variety of conditions encountered in the outdoor testing. Unfortunately, little data was available for wind speeds greater than 1 m/s.

Figure 9 presents measurements for a clear still night, as witnessed by the relatively low sky temperature and no wind. Figure 10 shows an

Table 2. Summary of Results from the Heat Loss Tests

Test Type	Date	Duration (h)	Initial Temp (C)	Final Temp (C)	Avg. Wind Speed (m/s)	Avg. Sky Temp (C)	Avg. Ambient Temp (C)	Lost Energy (MJ)	Overall Loss Coefficient (W/m^2-C)
OF	09/05/85	8	53.5	52.0	0.2	9.2	19.3	0.74	0.49
O	09/09/85	8	58.4	50.4	0.2	10.0	21.1	3.92	2.65
O	09/10/85	8	59.8	51.6	0.7	11.5	20.8	4.02	2.59
O	09/13/85	8	69.2	55.7	0.2	-14.6	5.2	6.60	2.59
O	09/14/85	8	71.1	57.5	0.1	-13.3	3.9	6.64	2.47
O	09/15/85	8	46.3	38.4	0.0	-10.1	6.4	3.89	2.43
O	09/16/85	8	56.2	46.6	0.0	- 8.3	7.7	4.71	2.43
O	09/17/85	8	55.1	45.8	0.0	- 5.5	9.2	4.57	2.49
O	09/18/85	8	55.4	46.4	0.0	- 2.7	10.9	4.42	2.48
O	09/19/85	8	55.0	45.8	0.0	- 0.7	11.7	4.52	2.63
O	09/20/85	8	52.5	44.3	0.0	0.2	12.1	4.03	2.50
O	09/21/85	8	48.4	42.1	0.1	8.1	15.4	3.10	2.34
O	09/22/85	8	45.7	39.5	0.0	6.3	14.1	3.05	2.41
O	09/24/85	8	45.0	38.9	0.3	- 5.5	9.6	3.01	2.09
O	09/25/85	8	47.8	41.5	0.4	11.3	14.8	3.10	2.34
OF	09/26/85	8	52.6	49.9	2.7	7.6	15.0	1.33	0.82
O	10/01/85	8	59.5	49.4	0.2	11.1	15.9	4.95	2.89
O	10/08/85	8	48.3	40.7	0.0	- 1.0	10.8	3.74	2.50
O	10/10/85	8	61.5	51.1	0.0	1.9	12.9	5.09	2.64
S	10/22/85	16	60.5	47.2	5.6	21.5	22.7	6.52	2.38
S	10/23/85	16	59.8	46.0	5.7	12.2	22.3	6.77	2.52
S	10/24/85	16	61.3	46.6	0.0	0.8	22.1	7.21	2.58
S	10/25/85	16	60.8	48.4	0.0	22.6	21.4	6.08	2.08
S	10/28/85	16	62.3	50.3	0.0	30.9	21.2	5.88	1.89
S	10/31/85	16	63.1	49.4	4.3	28.8	24.7	6.71	2.42
S	11/11/85	16	59.9	46.0	5.5	12.7	22.7	6.82	2.57
S	11/14/85	16	59.9	46.8	0.0	12.7	22.7	6.42	2.38

I = Indoor Stratified Test
O = Outdoor Stratified Test
S = SRCC Type Test
IF = Indoor Forced Test
OF = Outdoor Forced Test

(cont'd.)

Table 2. (cont'd.)

Test Type	Date	Duration (h)	Initial Temp (C)	Final Temp (C)	Avg. Wind Speed (m/s)	Avg. Sky Temp (C)	Avg. Ambient Temp (C)	Lost Energy (MJ)	Overall Loss Coefficient (W/m^2-C)
IF	09/12/85	8	53.1	50.4	0.0	7.2	21.0	1.33	0.97
I	09/13/85	8	60.0	53.6	0.0	22.0	20.4	3.13	1.93
I	09/14/85	8	60.0	52.9	0.0	9.6	20.9	3.48	2.20
I	09/15/85	8	60.0	52.4	0.0	1.6	20.4	3.72	2.34
I	09/16/85	8	60.0	52.0	0.0	- 6.4	19.8	3.92	2.44
I	09/17/85	8	60.0	52.3	2.3	4.8	19.9	3.77	2.34
I	09/18/85	8	60.0	52.4	2.2	5.9	20.0	3.72	2.31
I	09/19/85	8	60.0	52.4	2.2	8.6	19.9	3.72	2.31
I	09/20/85	8	60.0	52.7	2.2	19.0	20.1	3.58	2.22
IF	09/21/85	8	60.0	54.9	0.0	- 3.5	19.4	2.50	1.47
I	09/23/85	8	83.9	70.9	0.0	0.2	20.5	6.31	2.50
I	09/24/85	8	87.2	73.4	0.0	2.5	21.3	6.69	2.55
I	09/25/85	8	59.9	52.3	0.0	3.7	19.8	3.72	2.31
I	09/26/85	8	68.5	59.2	0.0	4.1	20.5	4.54	2.36
I	09/27/85	8	65.9	56.9	0.0	5.0	18.6	4.40	2.31
I	09/28/85	8	64.5	56.0	0.0	6.0	19.1	4.16	2.27
IF	11/03/85	8	54.3	51.0	0.3	10.2	21.3	1.62	1.16
I	11/04/85	8	53.5	47.7	0.0	16.2	21.5	2.85	2.20
I	11/05/85	8	53.1	47.2	0.0	12.6	21.2	2.90	2.25
I	11/07/85	8	55.6	49.1	0.4	5.0	22.3	3.19	2.39
I	11/12/85	8	59.9	52.4	0.9	11.3	21.3	3.67	2.37
I	11/13/85	8	58.3	51.0	0.2	9.9	20.8	3.58	2.38

evening when the sky turned from cloudy to clear with almost no wind. Figure 11 shows an example of an evening when the environmental parameters, T_s and V_w, varied considerably during the course of the test. In this particular test, the tank fluid was continuously mixed as evidenced by T_{top} equal to T_{bot}. In Figure 12, a fairly constant cloudy evening is presented with a wind speed that increased near the end of the test.

Figure 13 is a plot of the same test as Figure 11, except the data was plotted for five-minute intervals to show the significant variation of T_s, T_a, and V_w, within minutes. By comparing this figure to Figure 11, the change in the environmental parameters is shown to be greatly smoothed due to the hourly averaging.

When duplicating the outdoor tests indoors, the indoor environmental parameters were determined based on the eight-hour average of the outdoor parameters. By examining Figure 9 to Figure 13, it can be seen that some error can be introduced by this assumption.

As mentioned earlier, it is difficult to compare different outdoor tests results because of the variation in weather conditions. To emphasize this fact, Figure 14 and Figure 15 are presented. Figure 14 is a plot of several outdoor tests which have an ambient temperature within ±2 C of each other, an initial tank temperature within ±5 C of each other, and a wind speed of approximately zero. It can be seen that an increase in tank heat loss, as shown by a more rapid decrease in tank temperature, is consistent with a decrease in sky temperature from 8.1 C to -0.7 C. Specifically, a change in sky temperature of about 9 C in the absence of wind corresponds to a 31.4 per cent change in heat loss.

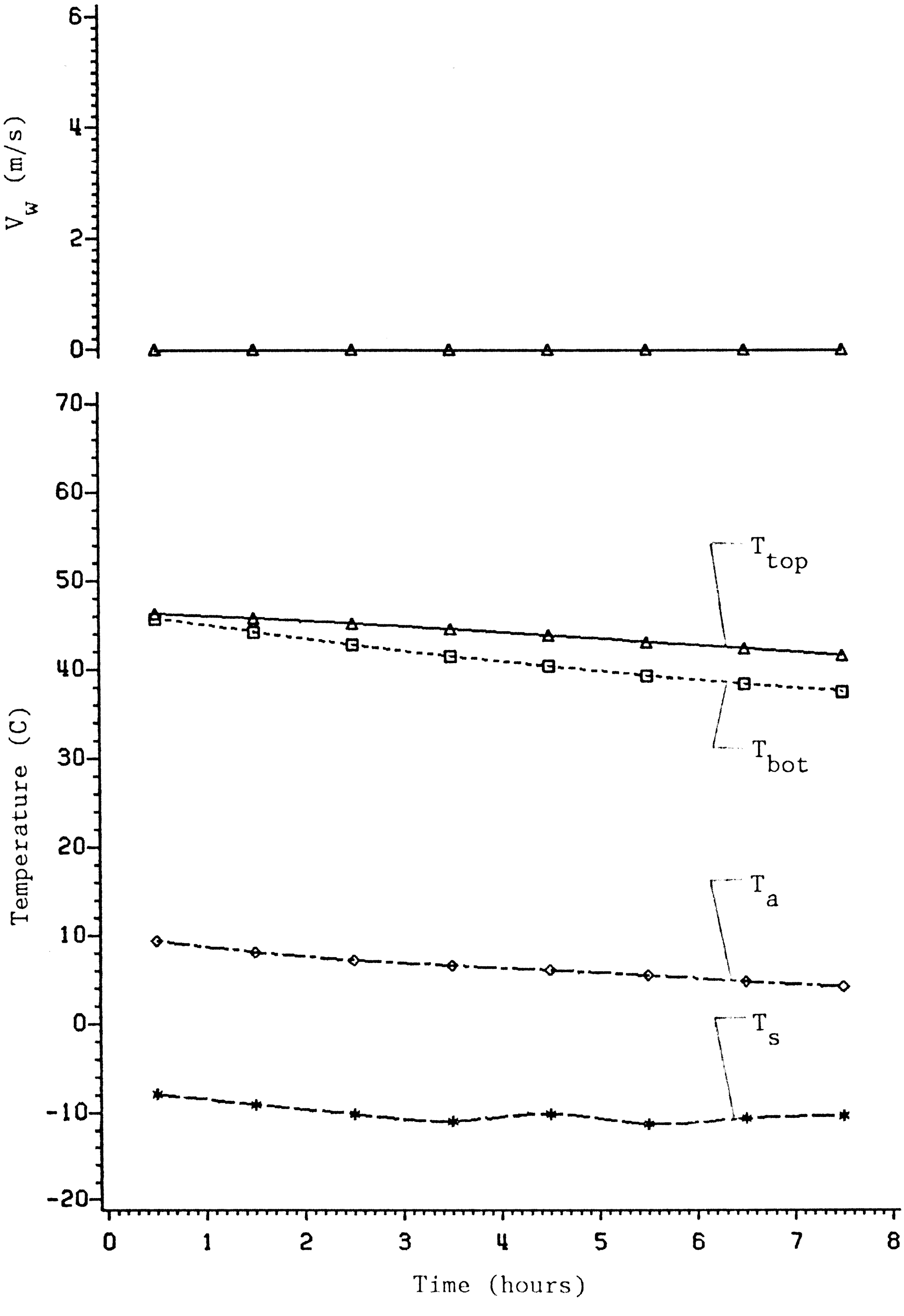

Figure 9. Outdoor Heat Loss Test 9/15/85 with Hourly Averaged Results

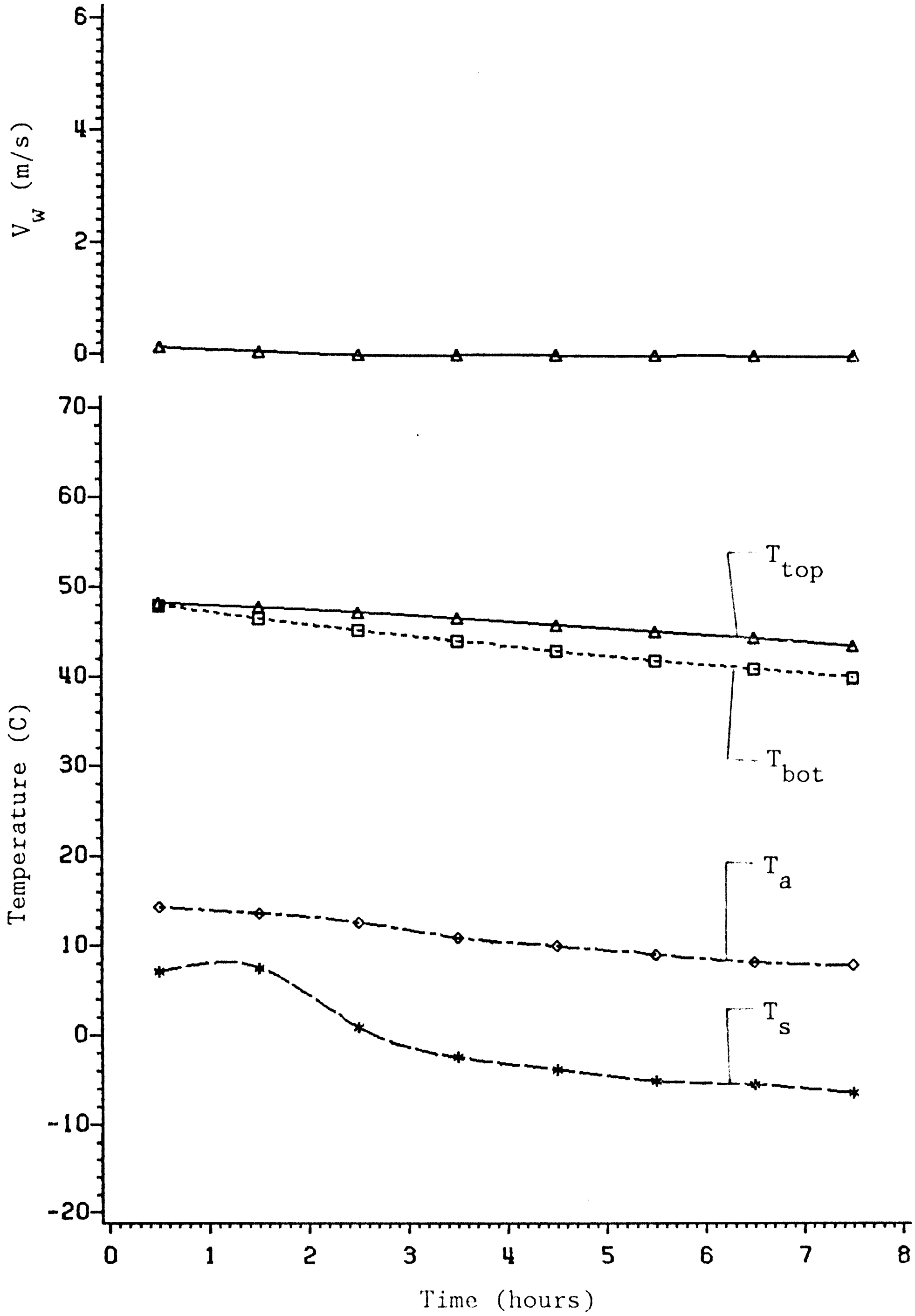

Figure 10. Outdoor Heat Loss Test 10/08/85 with Hourly Averaged Results

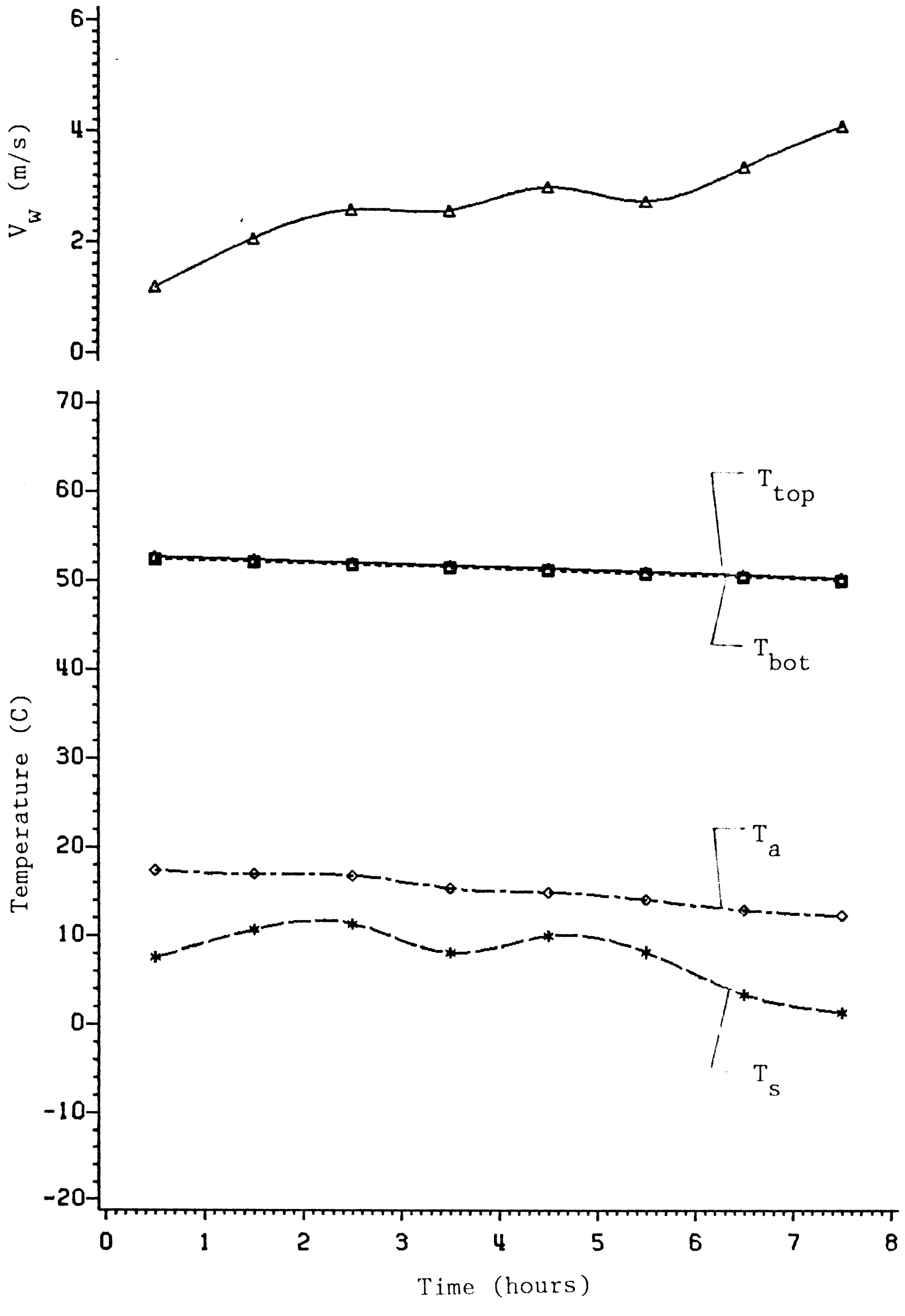

Figure 11. Outdoor Heat Loss Test 9/26/85 with Hourly Averaged Results

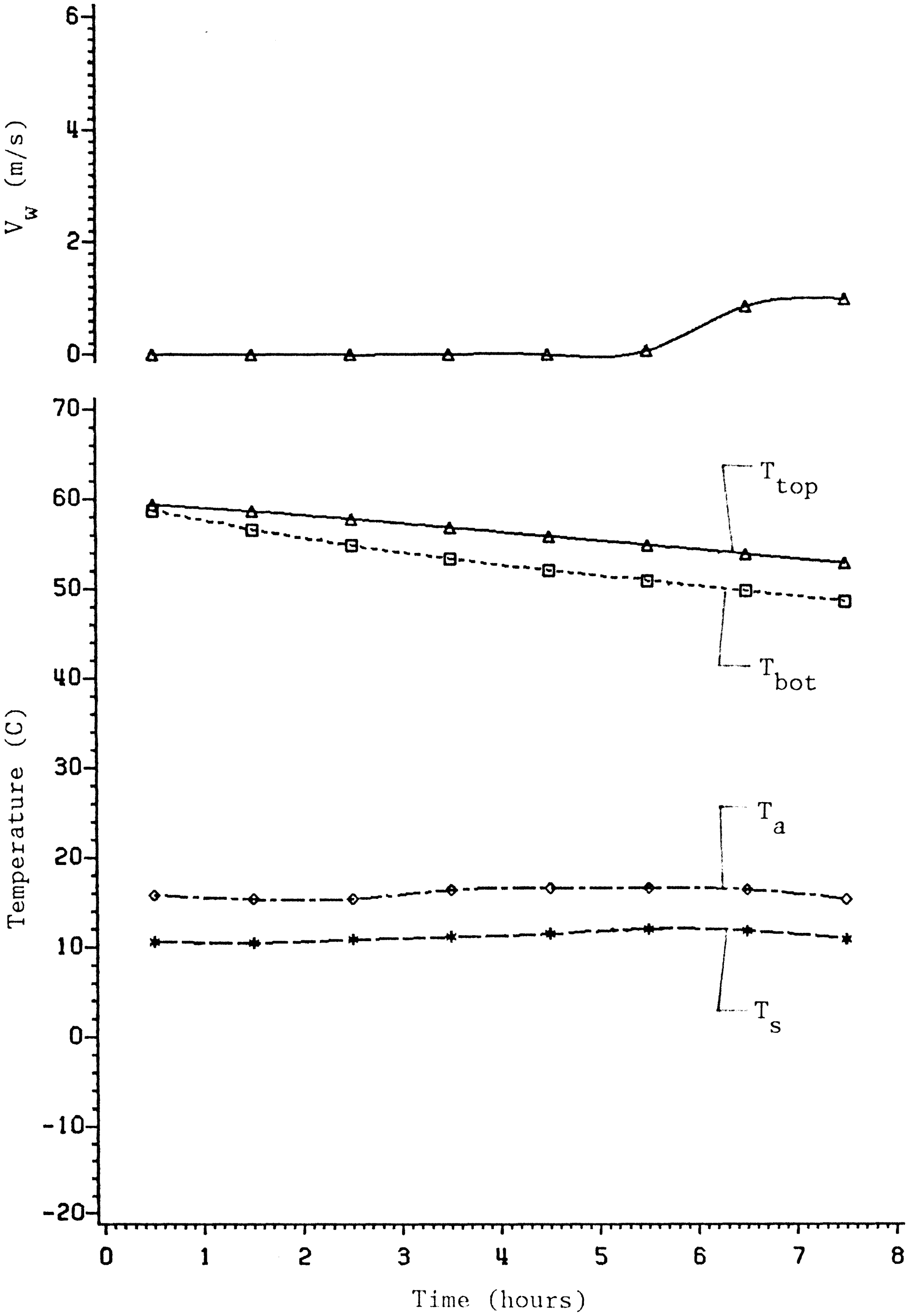

Figure 12. Outdoor Heat Loss Test 10/01/85 with Hourly Averaged Results

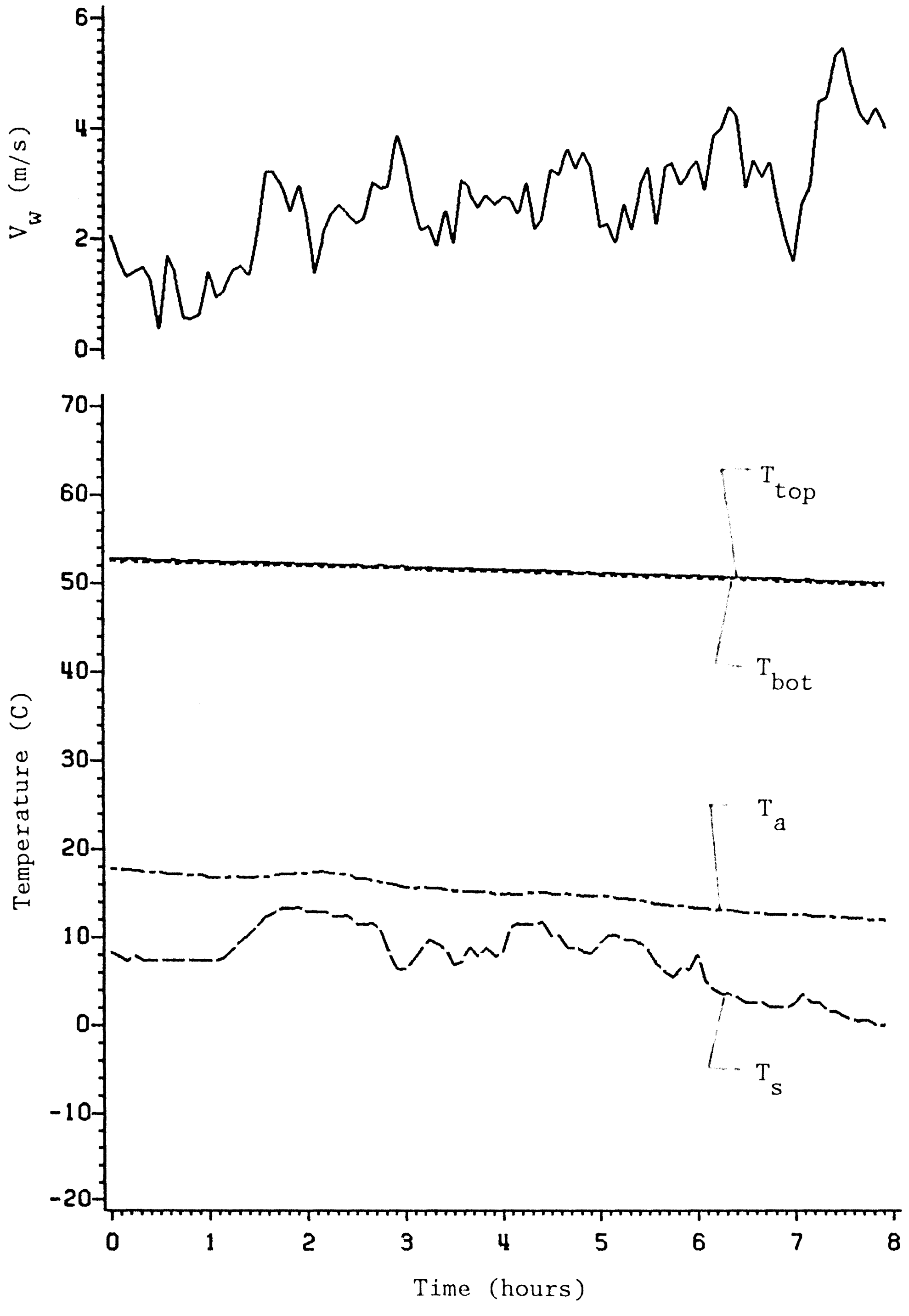

Figure 13. Outdoor Heat Loss Test 9/26/85 with 5-Minute Averaged Results

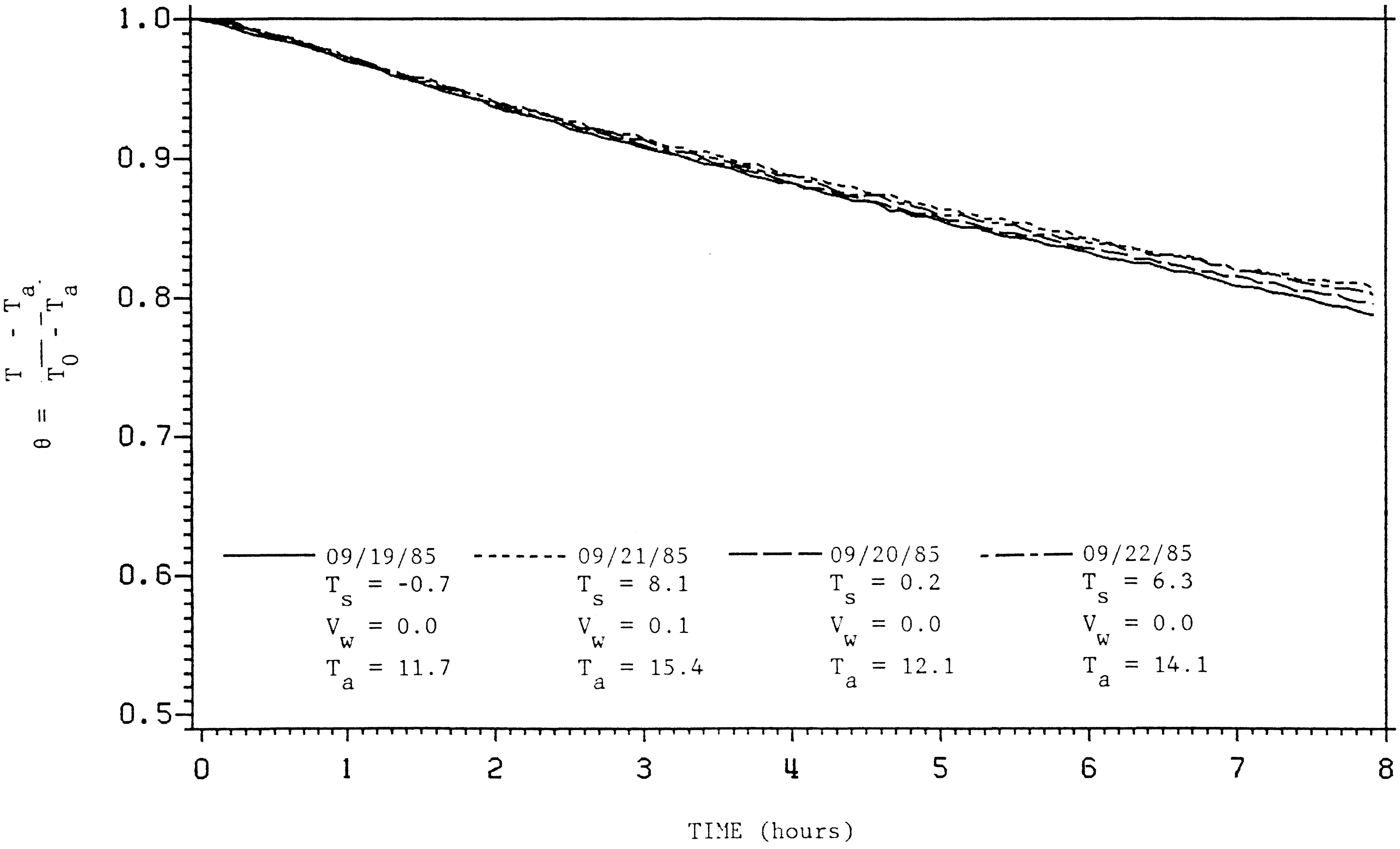

Figure 14. Comparison of Results from Outdoor Heat Loss Tests 09/19/85, 09/21/85, 09/20/85, 09/22/85

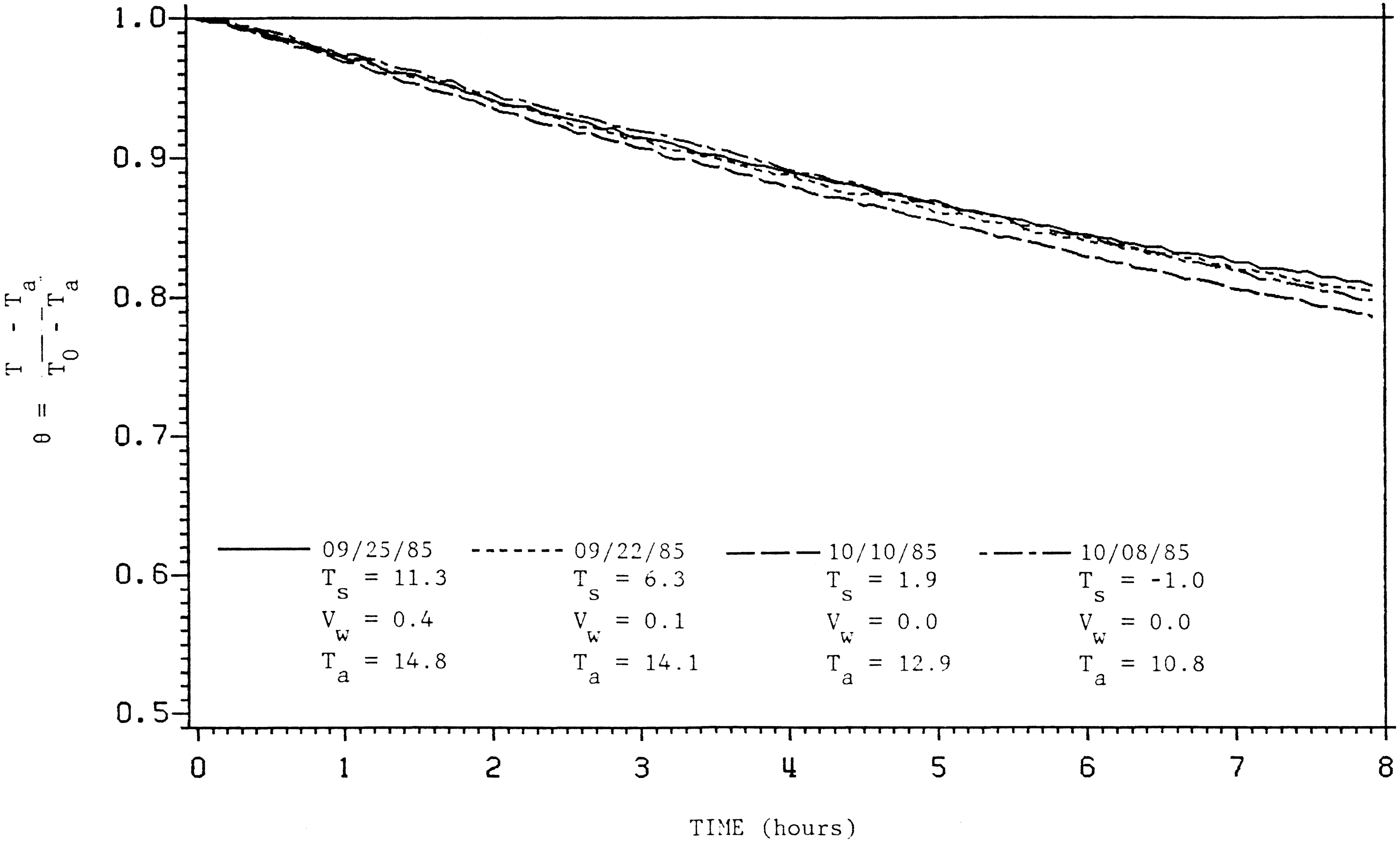

Figure 15. Comparison of Results from Outdoor Heat Loss Tests 09/25/85, 09/22/85, 10/10/85, 10/08/85

Figure 15 presents results for various outdoor tests with wind speeds close to zero, ambient temperatures within ±2 C and a range of sky temperatures. The final results however, are not as one might expect. The 10/08/85 test, with a sky temperature 3 C lower than the 10/10/85 test, resulted in a smaller amount of heat loss than the 10/10/85 case. As can be seen in Appendix C, the initial tank temperatures are quite a bit different. As one might expect, the 10/10/85 test had a higher initial temperature. This therefore emphasizes the need to conduct tests in which certain parameters can be controlled so that accurate conclusions can be deduced. It is also important to note the synergistic effect between the different environmental parameters. For example, a small amount of wind may reverse the expected trend of sky temperature or changes in ambient temperature may overshadow the effects of changes in the sky temperature. It is this type of effect or combination of effects that is being examined here.

4.2 COMPARISON OF INDOOR TESTS AND OUTDOOR TESTS

4.2.1 METHOD OF COMPARISON

Since heat loss tests for rating purposes are currently conducted indoors, it was desired to see how well an outdoor test conducted at night, under prevailing weather conditions, could be simulated indoors. In order to duplicate the outdoor thermal performance, it was necessary to perform the indoor tests under the same environmental conditions and initial tank temperature as occurred outdoors. While the sky temperature, wind speed, and initial tank temperature were not difficult to duplicate, the lower ambient air temperatures presented a problem. A method was devised to calculate equivalent indoor parameters which would have the same difference as the outdoor parameters. Values for X and Y in equations (11) and (12) were calculated based on the T_0, T_s, and T_a values obtained outdoors.

$$X = T_0 - T_s \tag{11}$$

$$Y = T_a - T_s \tag{12}$$

To obtain the equivalent indoor parameters a new value of T_a=20 C was substituted into equation (12) to determine a new value of T_s and T_0. All environmental parameters for the indoor tests were set based on a T_a of 20 C, except the wind speed which was duplicated directly.

To better analyze the results of this section, a dimensionless heat loss parameter, R, was defined as

$$R = \frac{(\theta_0 - \theta_f)_{in}}{(\theta_0 - \theta_f)_{out}} \tag{13}$$

This parameter served as a basis of comparison. If R was less than 1.0, then the indoor model underpredicted the heat loss, and if R was greater than 1.0, then the indoor model overpredicted the losses.

4.2.2 RESULTS

Results from the indoor-outdoor comparison tests are presented in Figure 16 through Figure 26, as a function of the nondimensional temperature variable, θ, vs time. Average hourly data from the tests can be found in Appendix C and the results are also summarized in Table 2.

In Figure 16, the outdoor test shows a greater decrease in temperature during the first 3 hours while the indoor test is almost a uniform decrease over the 8-hour period. Although the two curves vary a bit in the center, the heat loss values are very close, with R=0.97. Figure 17 shows similar results to Figure 16 but at the end of the test the two curves crossed. This was due to the fact that the end of the test extended a bit into daylight giving a value of R=1.03. In both outdoor tests of Figure 16 and Figure 17, there was a small amount of wind but it was not possible to duplicate because it was too small.

Figure 18 and Figure 19, both for no wind and a sky temperature near 4 C, show fairly consistent results between the indoor and outdoor tests. Both plots yield values of R=0.96.

Figure 20 and Figure 21, for a slightly higher sky temperatures than the tests in Figure 18 and Figure 19, show a fairly good comparison between the indoor and outdoor results, yielding values of R=0.94 and R=0.93, respectively.

Figure 22 and Figure 23, for indoor sky temperatures of 12.6 C and 16.2 C, respectively, show a slight variation between the indoor and outdoor results during the middle of the test but they converge toward the end of the test. This trend could be due in part to the fluctuation

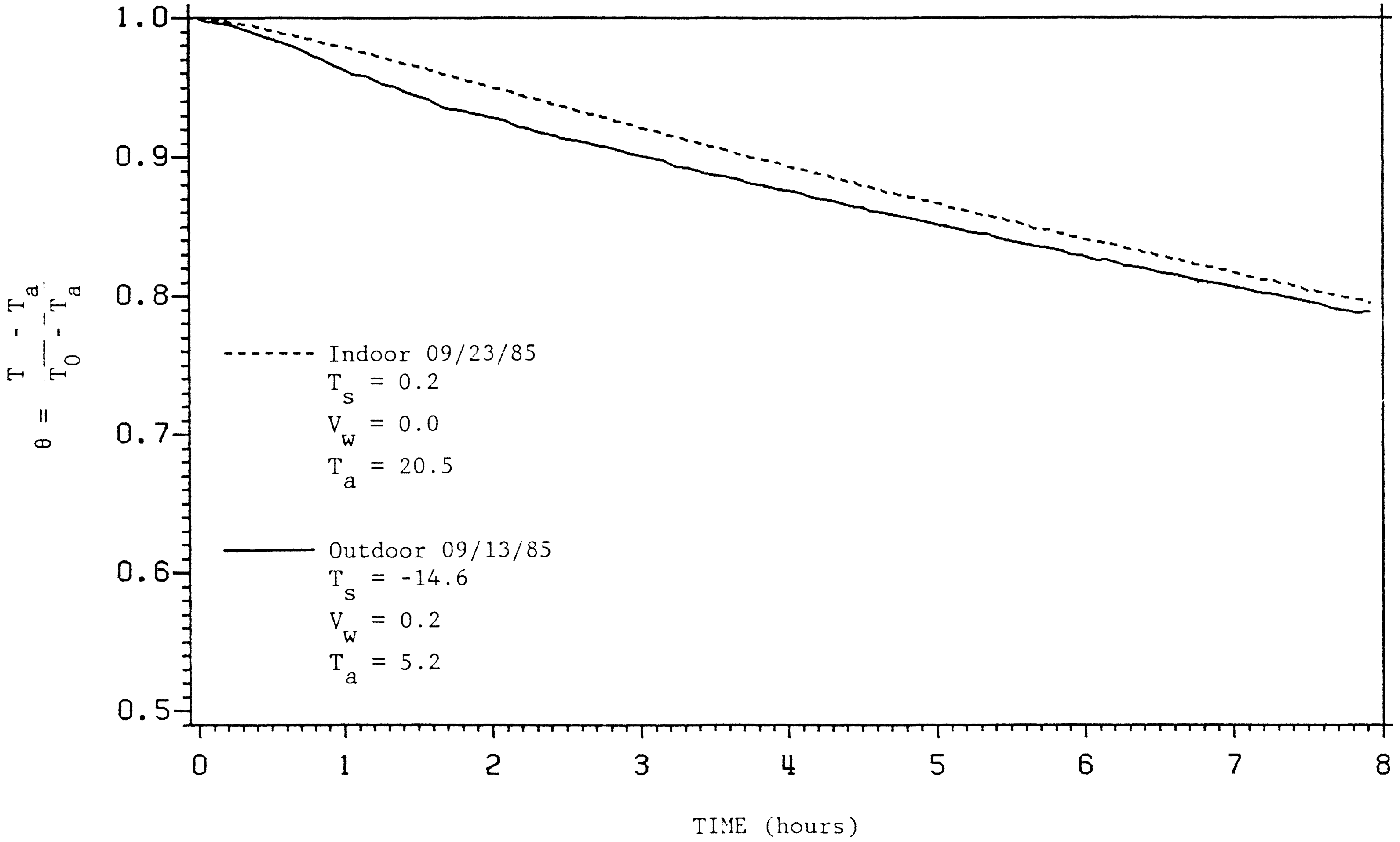

Figure 16. Comparison of Results from Indoor Heat Loss Test 09/23/85 and Outdoor Heat Loss Test 09/13/85

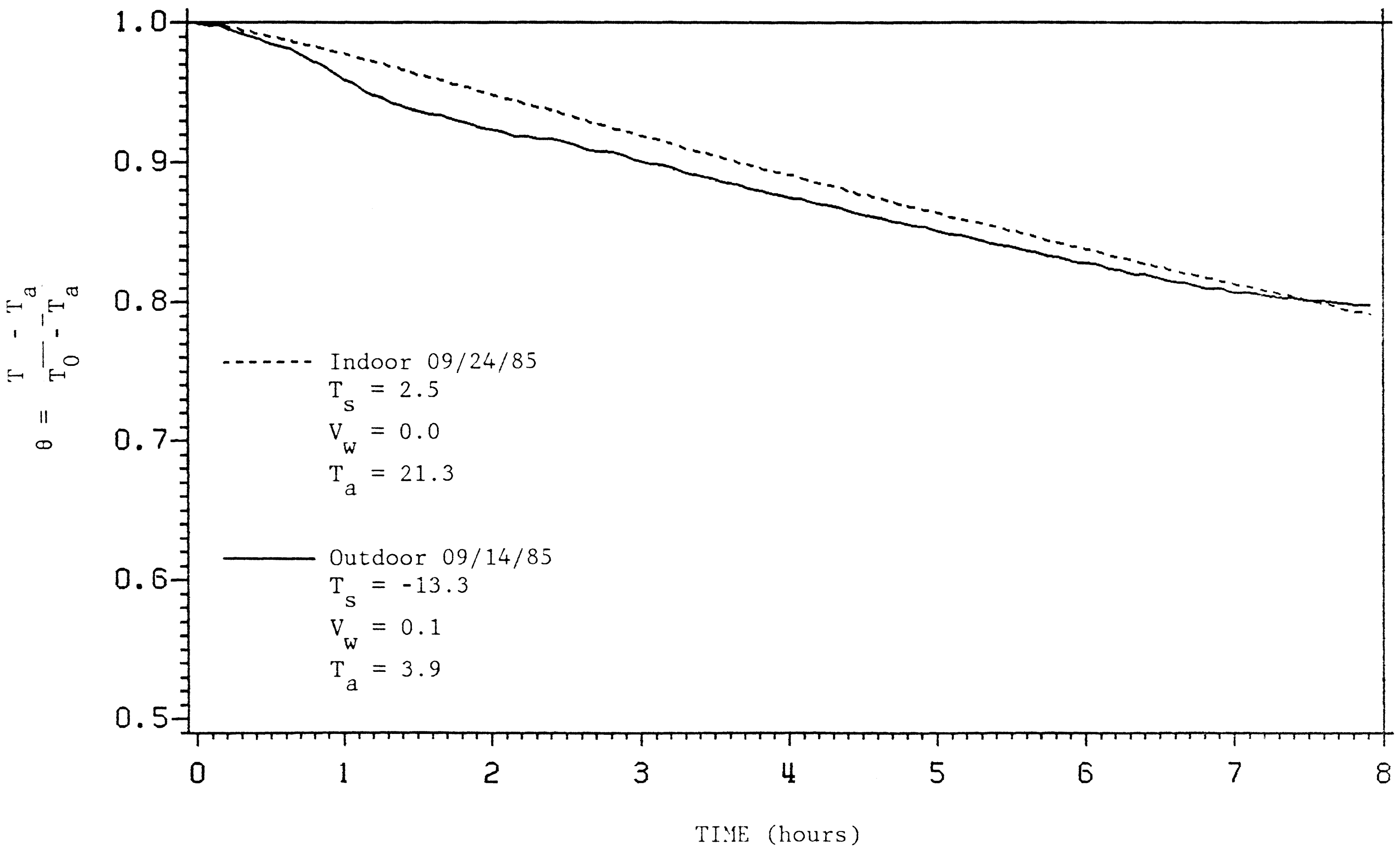

Figure 17. Comparison of Results from Indoor Heat Loss Test 09/24/85 and Outdoor Heat Loss Test 09/14/85

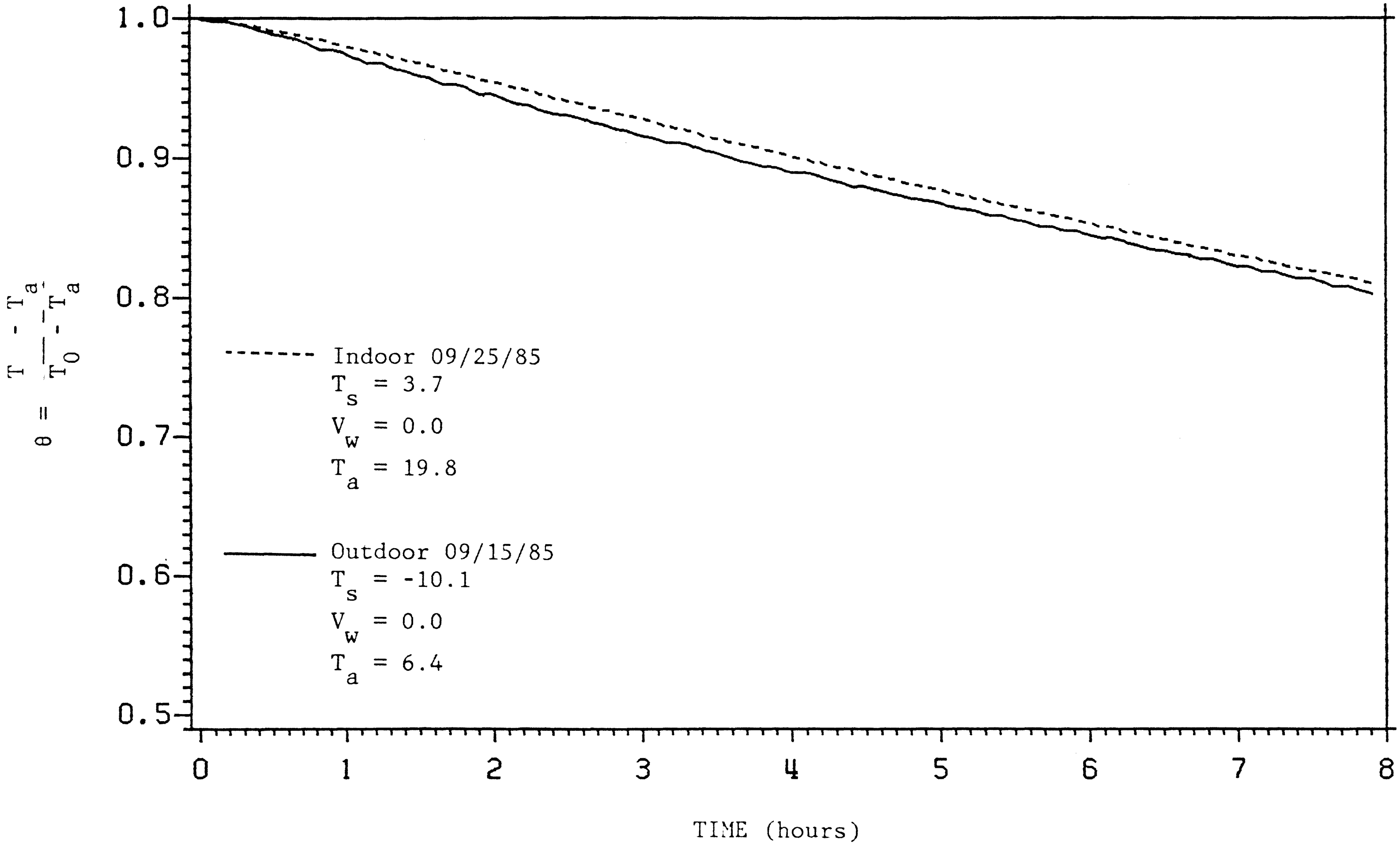

Figure 18. Comparison of Results from Indoor Heat Loss Test 09/25/85 and Outdoor Heat Loss Test 09/15/85

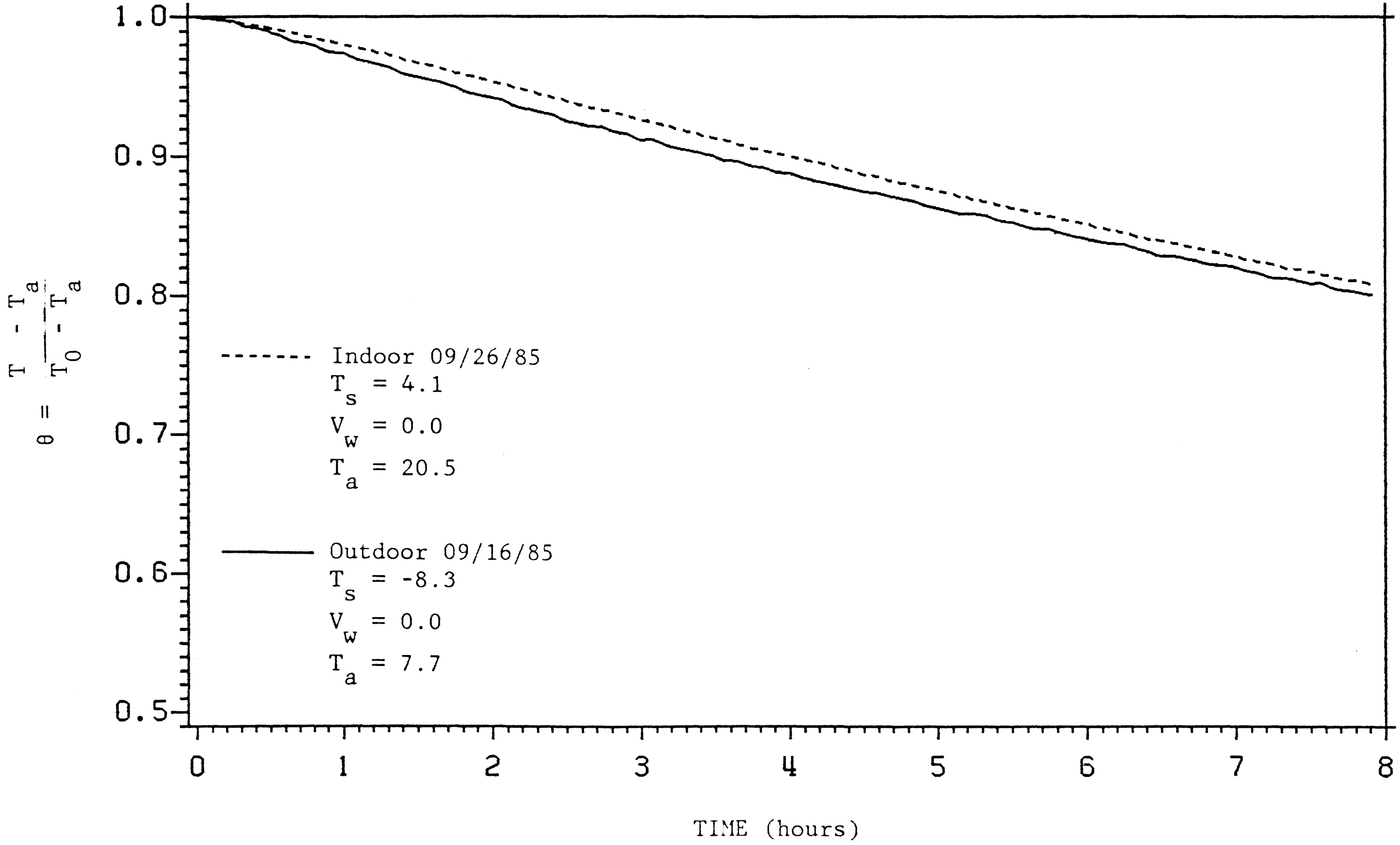

Figure 19. Comparison of Results from Indoor Heat Loss Test 09/26/85 and Outdoor Heat Loss Test 09/16/85

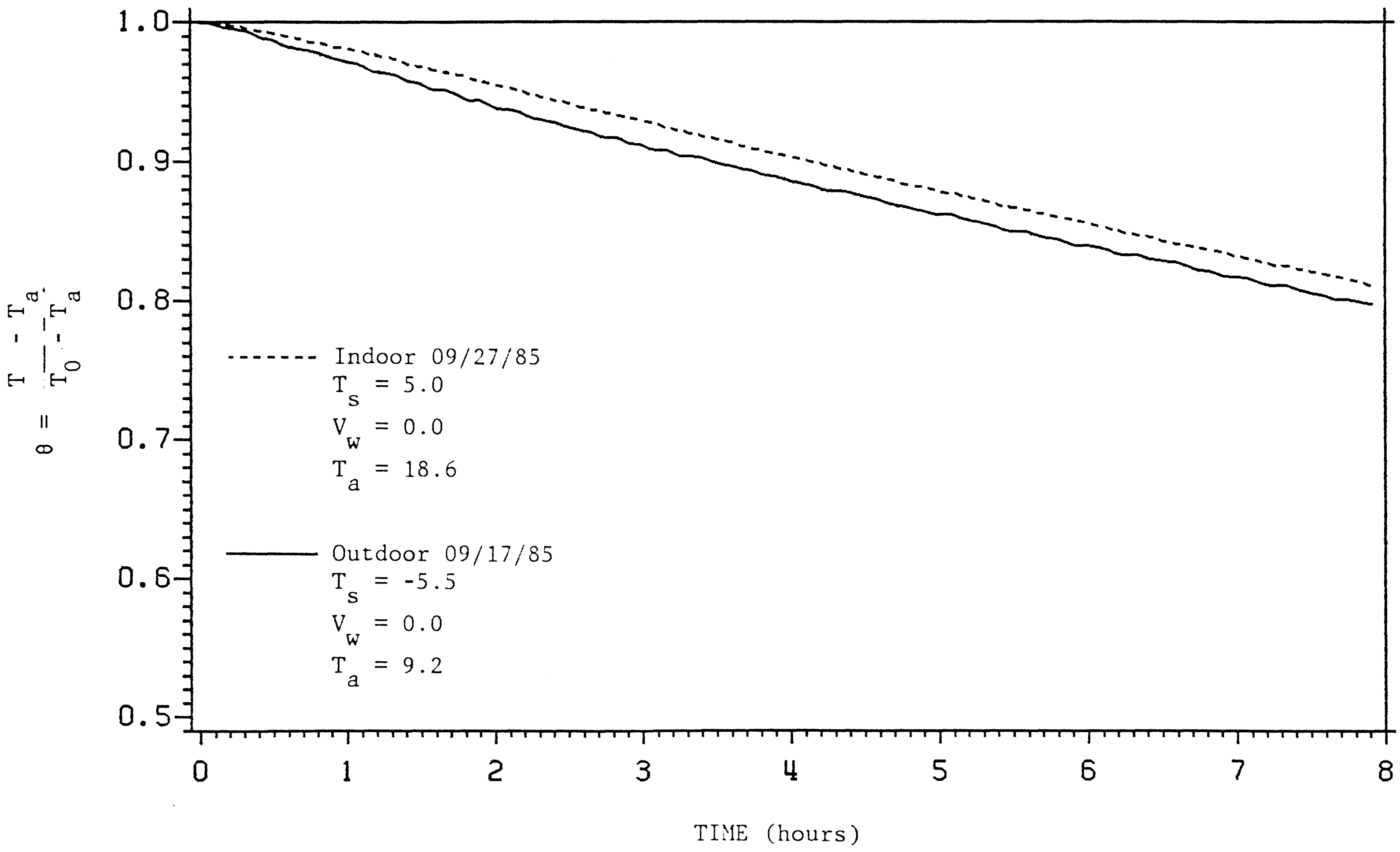

Figure 20. Comparison of Results from Indoor Heat Loss Test 09/27/85 and Outdoor Heat Loss Test 09/17/85

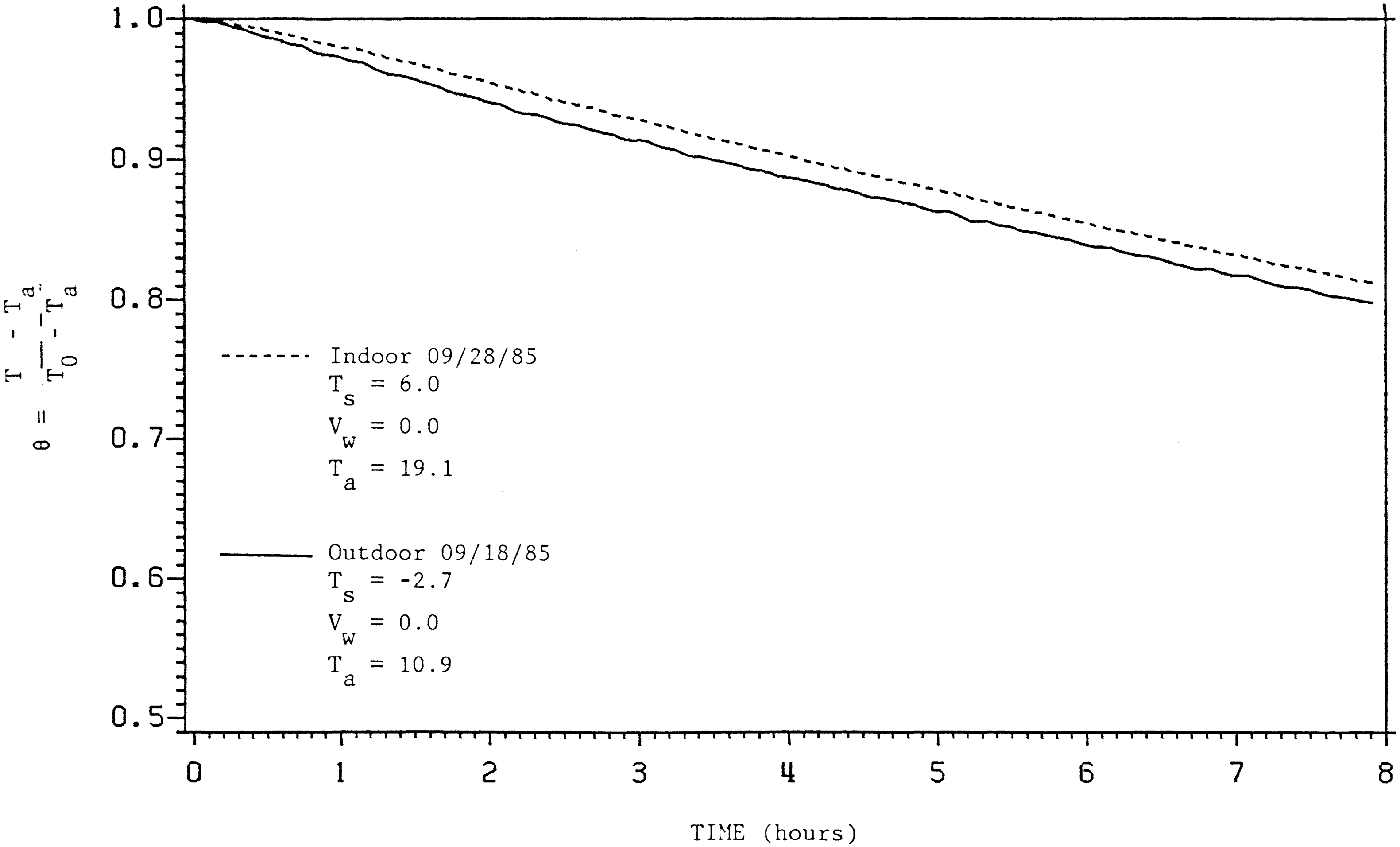

Figure 21. Comparison of Results from Indoor Heat Loss Test 09/28/85 and Outdoor Heat Loss Test 09/18/85

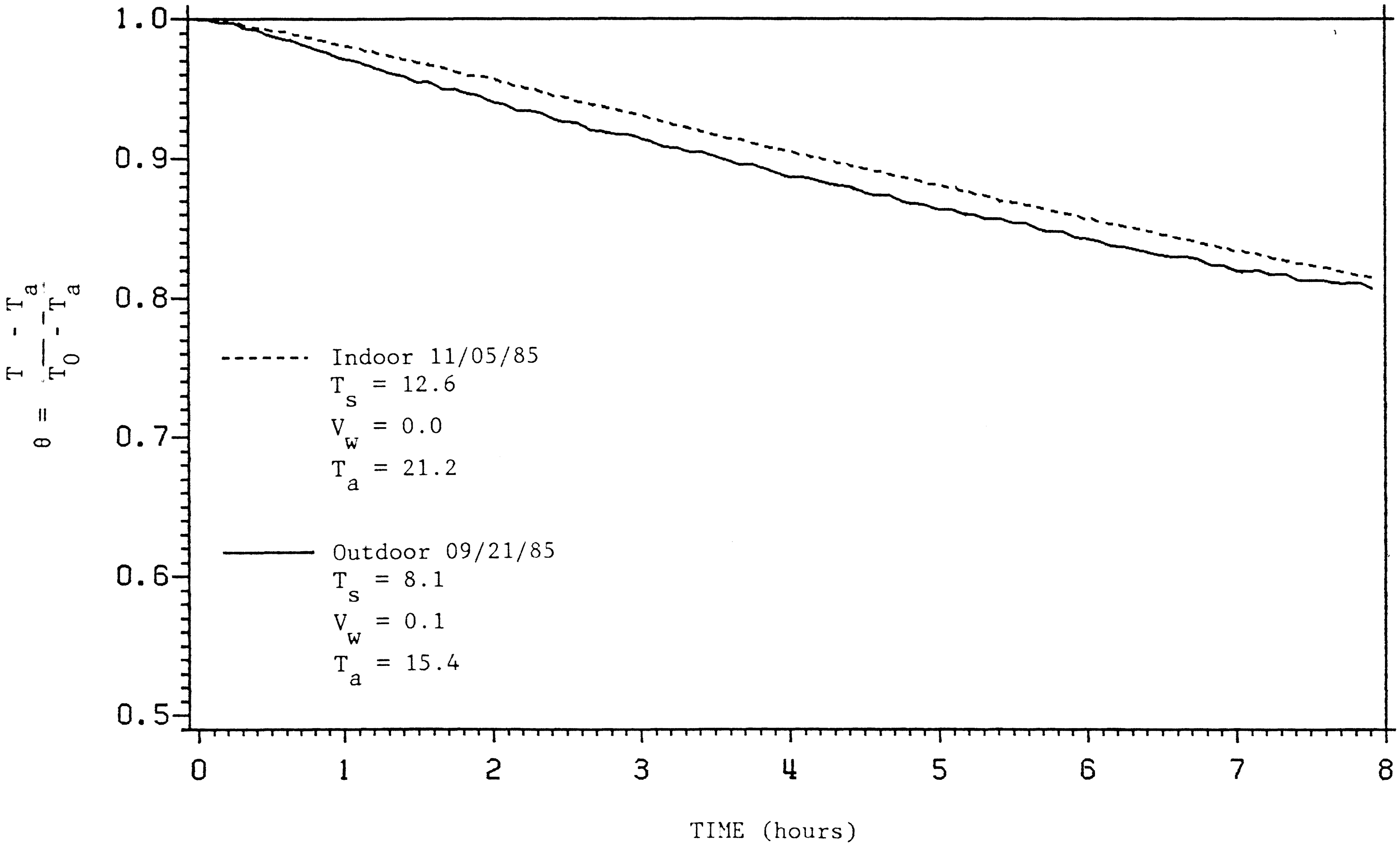

Figure 22. Comparison of Results from Indoor Heat Loss Test 11/05/85 and Outdoor Heat Loss Test 09/21/85

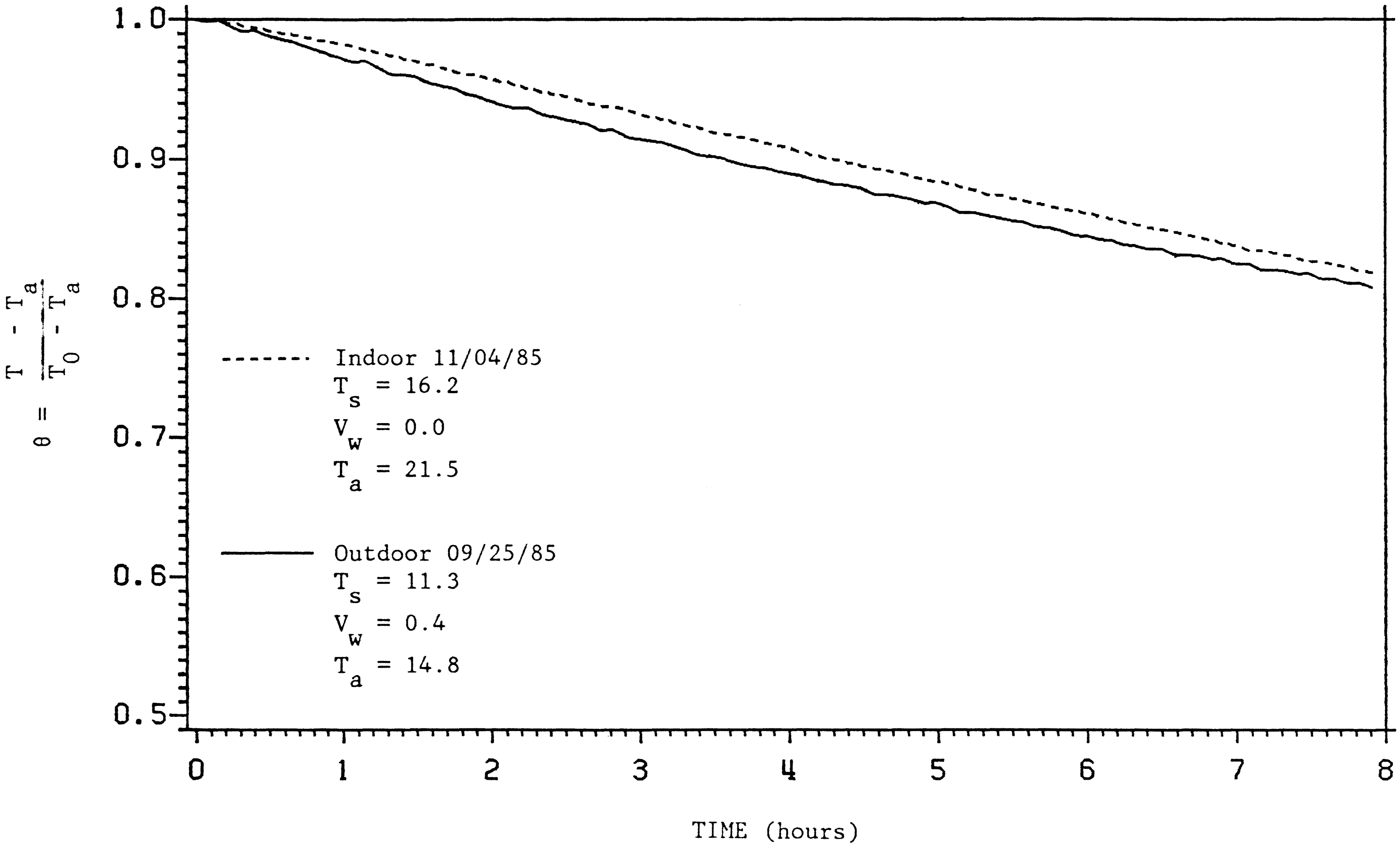

Figure 23. Comparison of Results from Indoor Heat Loss Test 11/04/85 and Outdoor Heat Loss Test 09/25/85

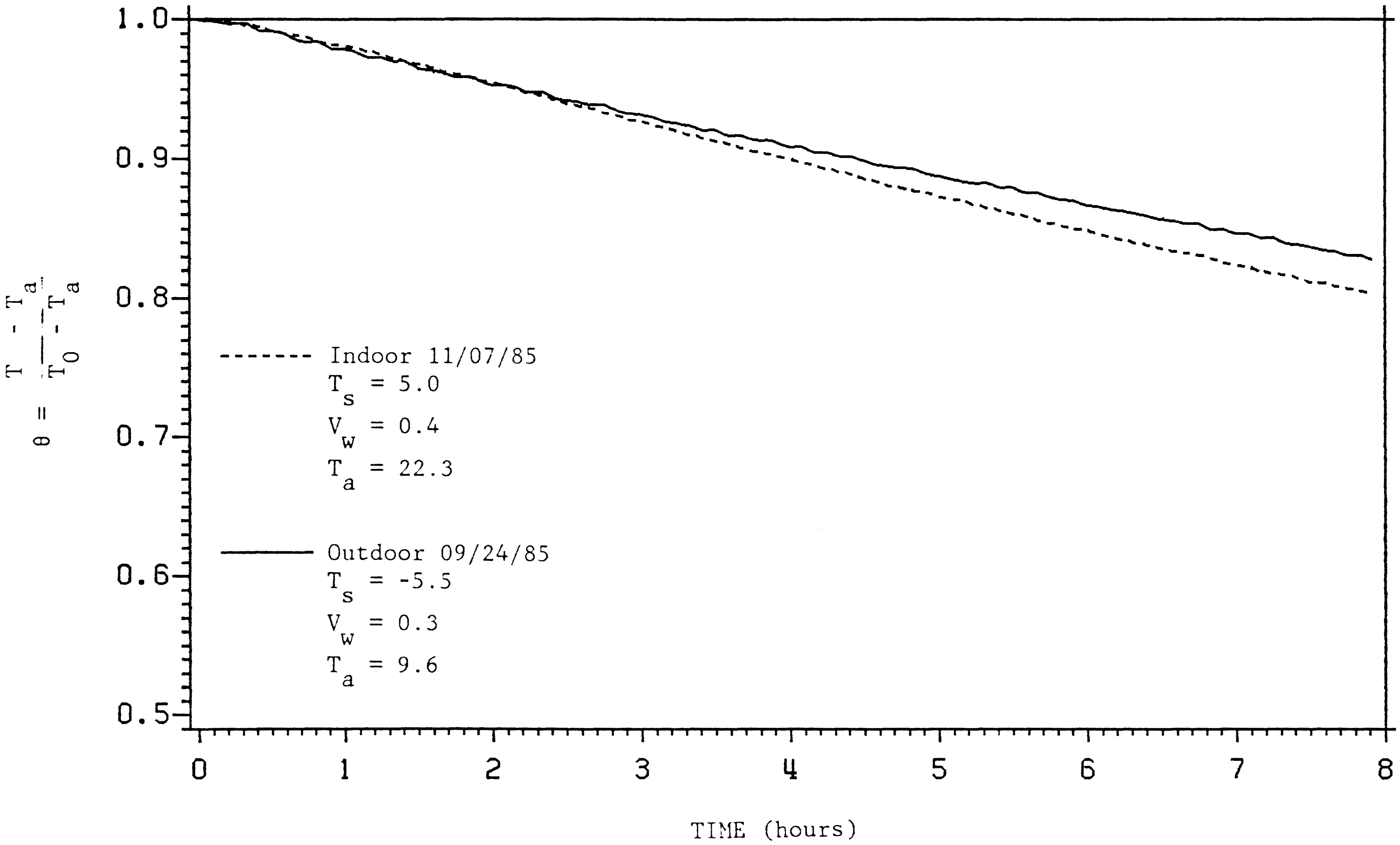

Figure 24. Comparison of Results from Indoor Heat Loss Test 11/07/85 and Outdoor Heat Loss Test 09/24/85

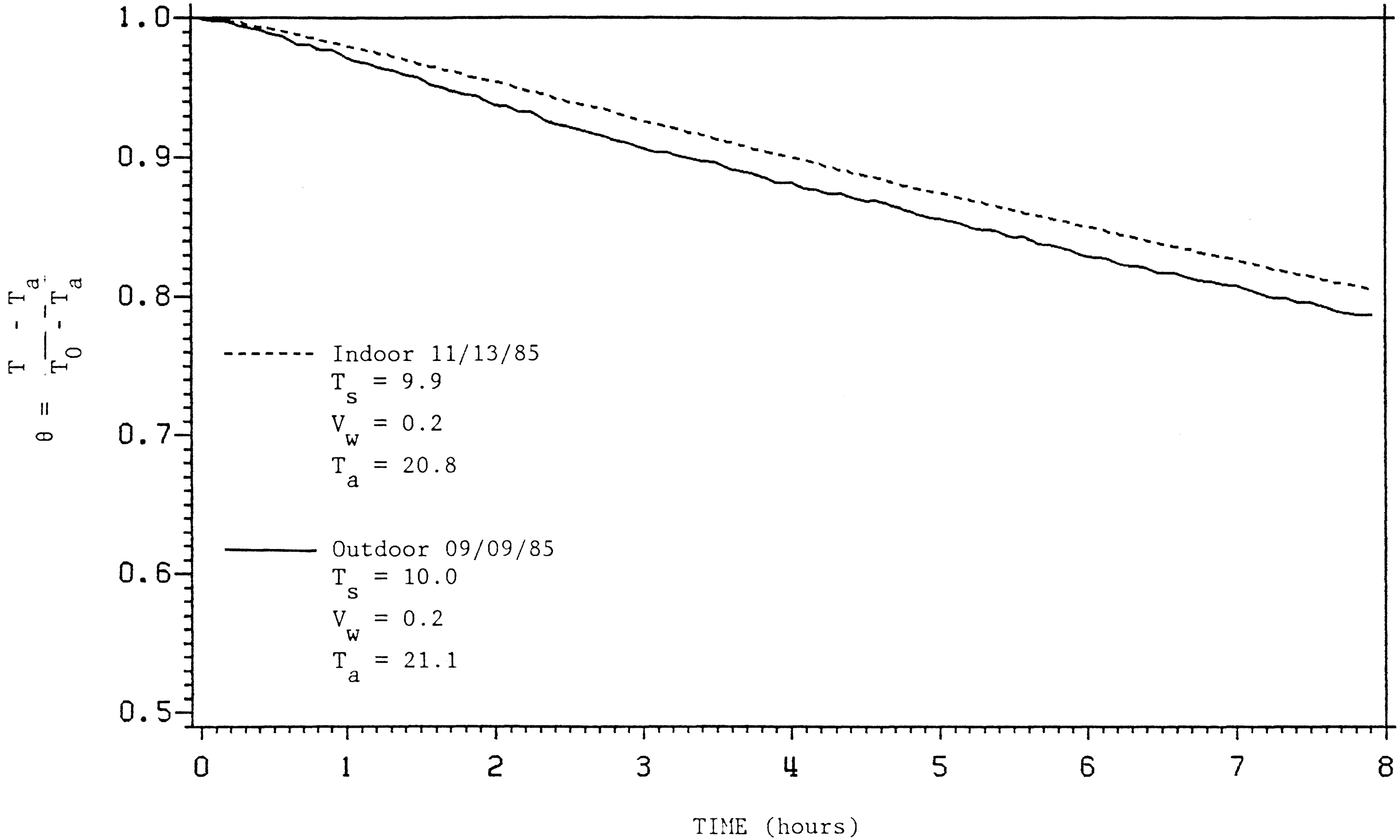

Figure 25. Comparison of Results from Indoor Heat Loss Test 11/13/85 and Outdoor Heat Loss Test 09/09/85

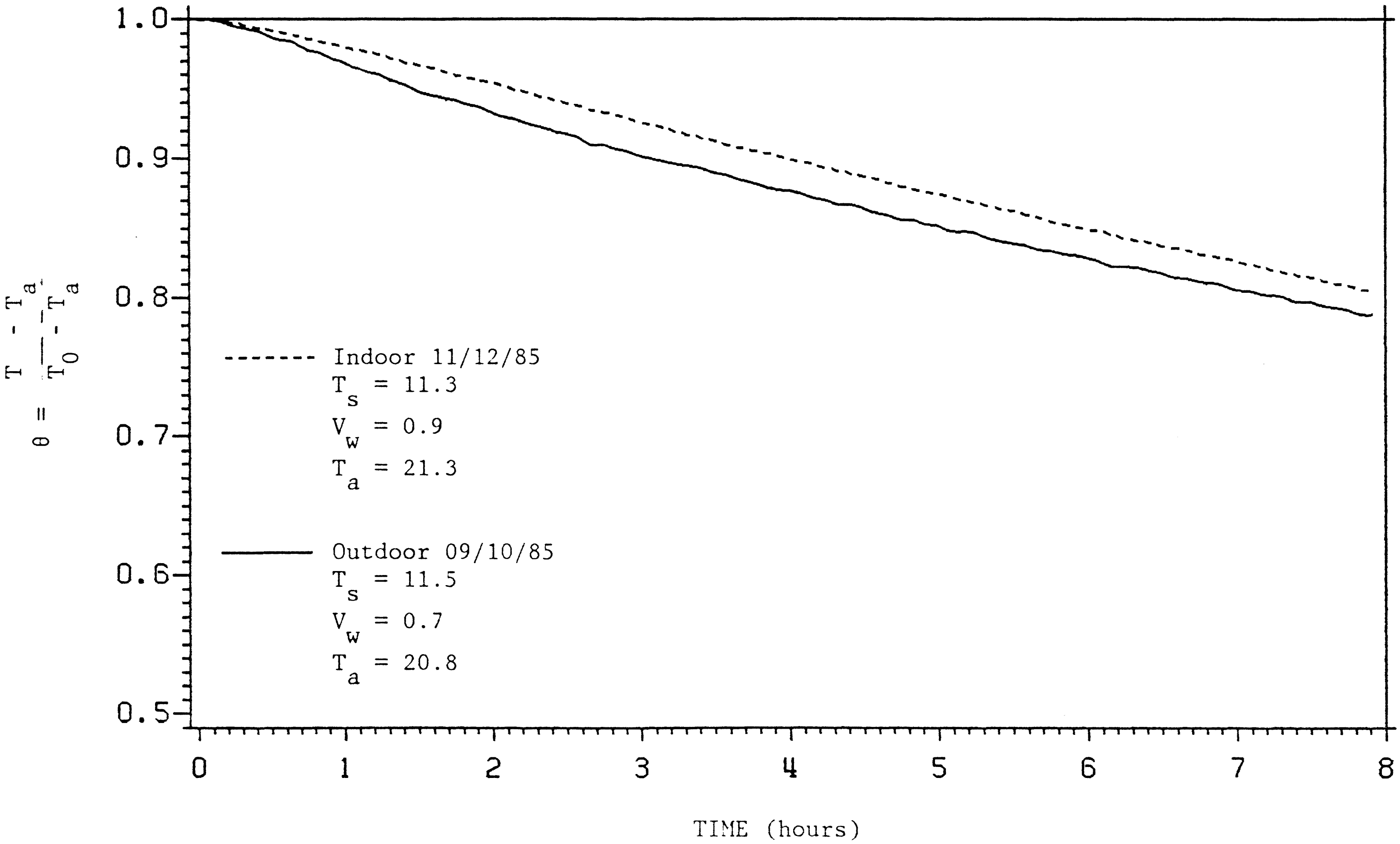

Figure 26. Comparison of Results from Indoor Heat Loss Test 11/12/85 and Outdoor Heat Loss Test 09/10/85

of the environmental parameters outdoors. However, a close review of the measured results did not provide additional information to support this claim. The outdoor test, 09/25/85, in Figure 23 actually had a wind speed of 0.40 m/s but no wind was supplied for the indoor test. Figure 22 yielded a value of R=0.96, and for Figure 23 a value of R=0.94 was calculated.

The results plotted in Figure 24 are interesting because this was the only case in the indoor-outdoor comparison tests when the final value of θ was greater for the outdoor test than for the indoor test. The two curves are very close together for about 2.5 hours and then the outdoor results diverge above the indoor test results. The value of the ratio of the heat losses is 1.15. After examining the outdoor experimental data it can be seen that while the sky temperature fluctuates about -5.5 C during the entire test, the majority of the wind is experienced within the first two hours of testing and then it dies down to zero. This case is an example of how using the hourly average values can lead to unexpected results.

Figure 25 and Figure 26 present outdoor cases with ambient temperatures that are very close to 20 C. The environmental parameters for the indoor tests were duplicated exactly as opposed to basing the parameters on an ambient temperature of 20 C as discussed in Section 4.2.1. The heat loss ratio for both cases was found to be 0.92. Although the test results still show good agreement, these two comparisons are the worst of those observed. If one were just to examine the outdoor ambient temperatures for all of the 10 indoor-outdoor comparison cases, it appears that the closest comparison of the heat loss occurs at the lower outdoor ambient temperatures which correspond to lower sky temperatures indoors.

Excluding the tests in Figure 24, all of the indoor heat loss test results were within 8 per cent of those for the corresponding outdoor tests. These observations were made over a range of outdoor parameters: the ambient temperature was between 3.9 C and 21.1 C, the sky temperature was between -14.6 C and 11.5 C, and the wind speed varied between 0 and 0.7 m/s. It is therefore concluded that outdoor heat loss test results can be simulated indoors with relatively good accuracy providing the environmental parameters are accounted for and modeled in a suitable manner.

4.3 INDOOR TESTS

Several tests were run indoors at the same initial tank temperature of 60 C and an ambient temperature of 20 C but with varied sky temperatures and wind speeds. The results for several tests were plotted on one figure to assess the relative effects of the sky temperature and wind speed. The tests ran for eight hours and were set up so that one parameter was varied while the others remained constant. Figure 27 through Figure 30 show the results from these indoor comparison tests with the nondimensional temperature θ, plotted against time. The results are based on experimental data taken at 5 minute intervals. Table 2 and Appendix C give further information about each of the tests conducted.

Figure 27 shows the relative effect of sky temperature with no wind. The results show a 20.2 per cent increase in heat loss for a 28 C decrease in sky temperature. In Figure 28, the effect of sky temperature is

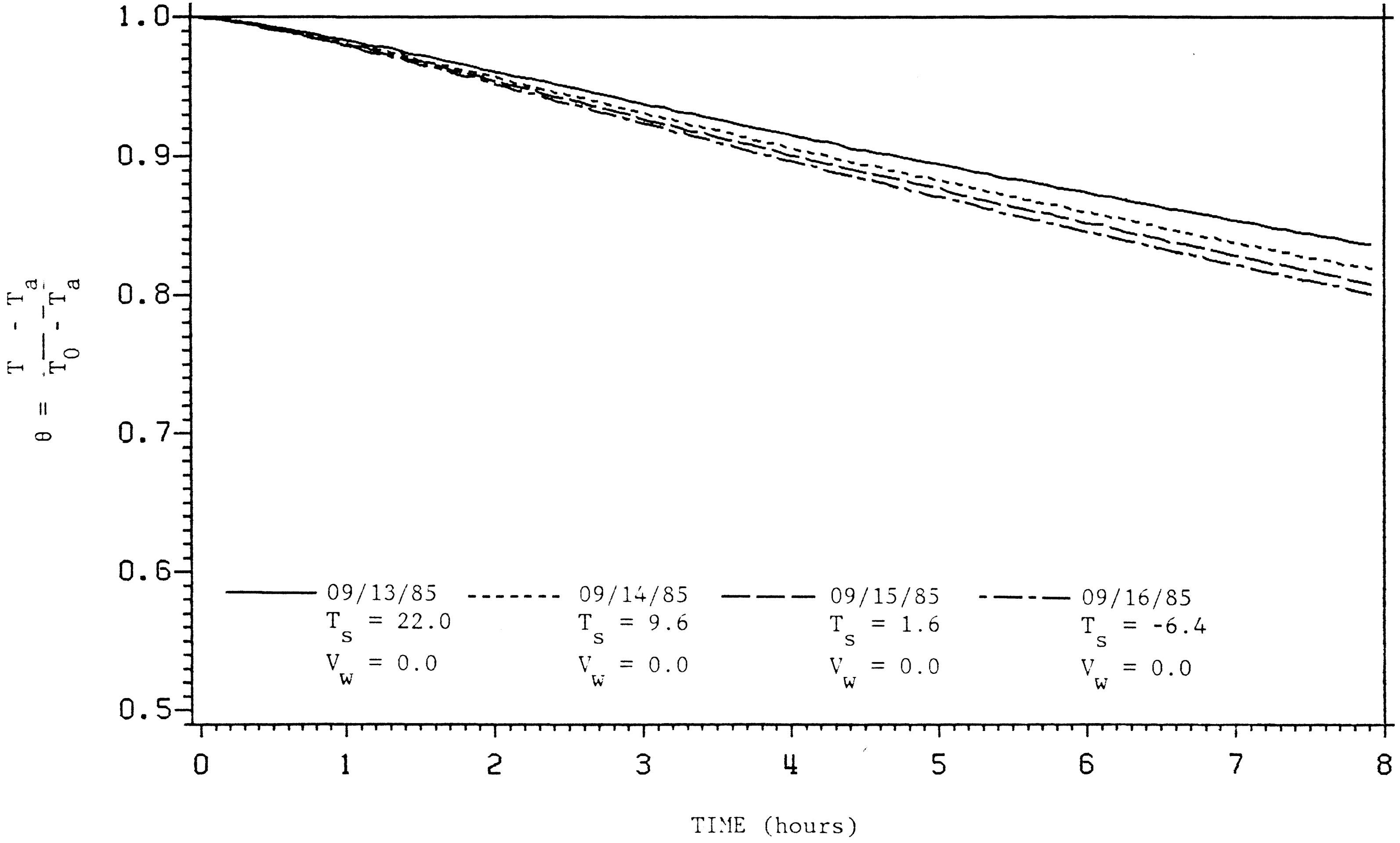

Figure 27. Comparison of Results from Indoor Heat Loss Tests 09/13/85, 09/14/85, 09/15/85, 09/16/85

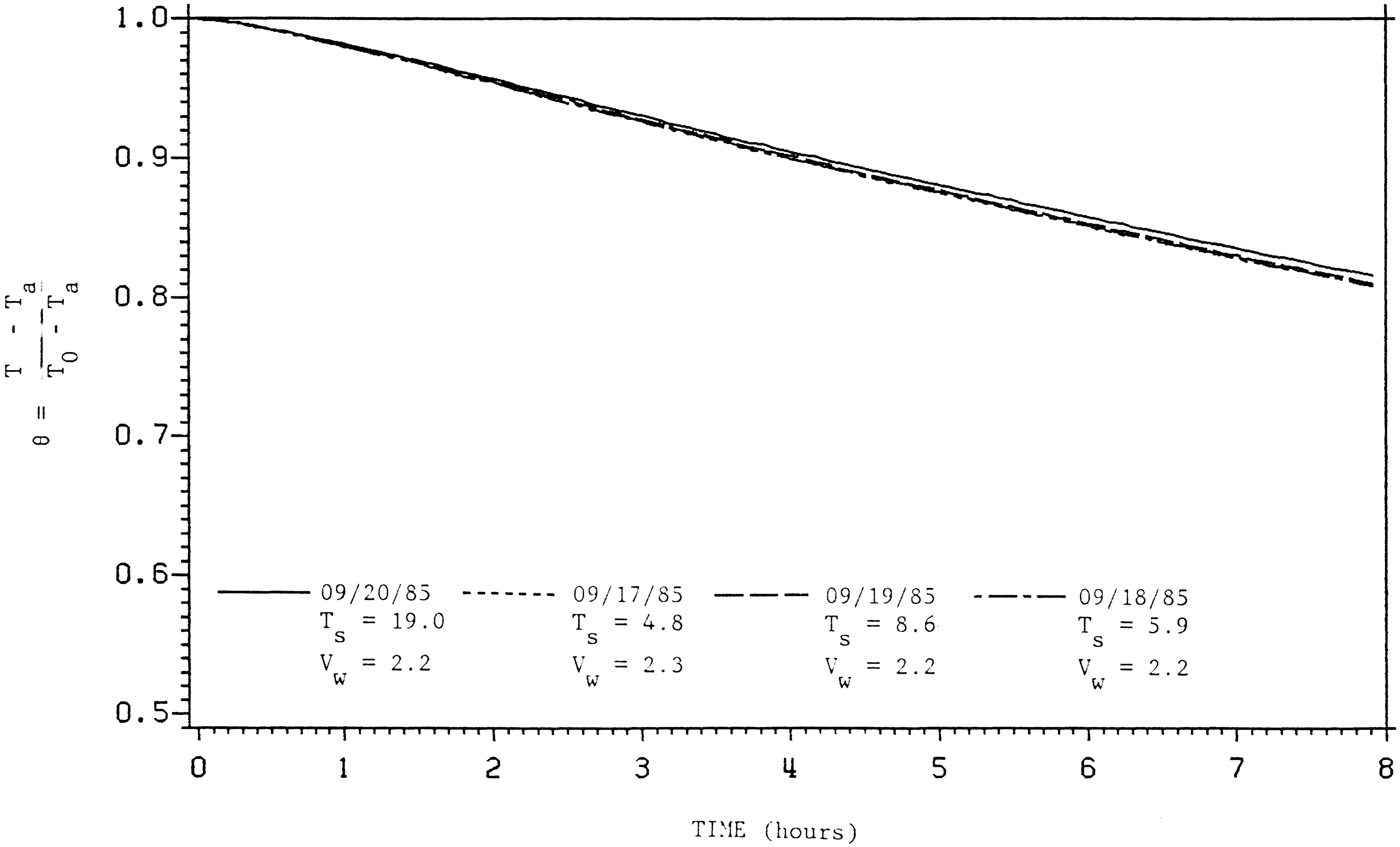

Figure 28. Comparison of Results from Indoor Heat Loss Tests 09/20/85, 09/17/85, 09/19/85, 09/18/85

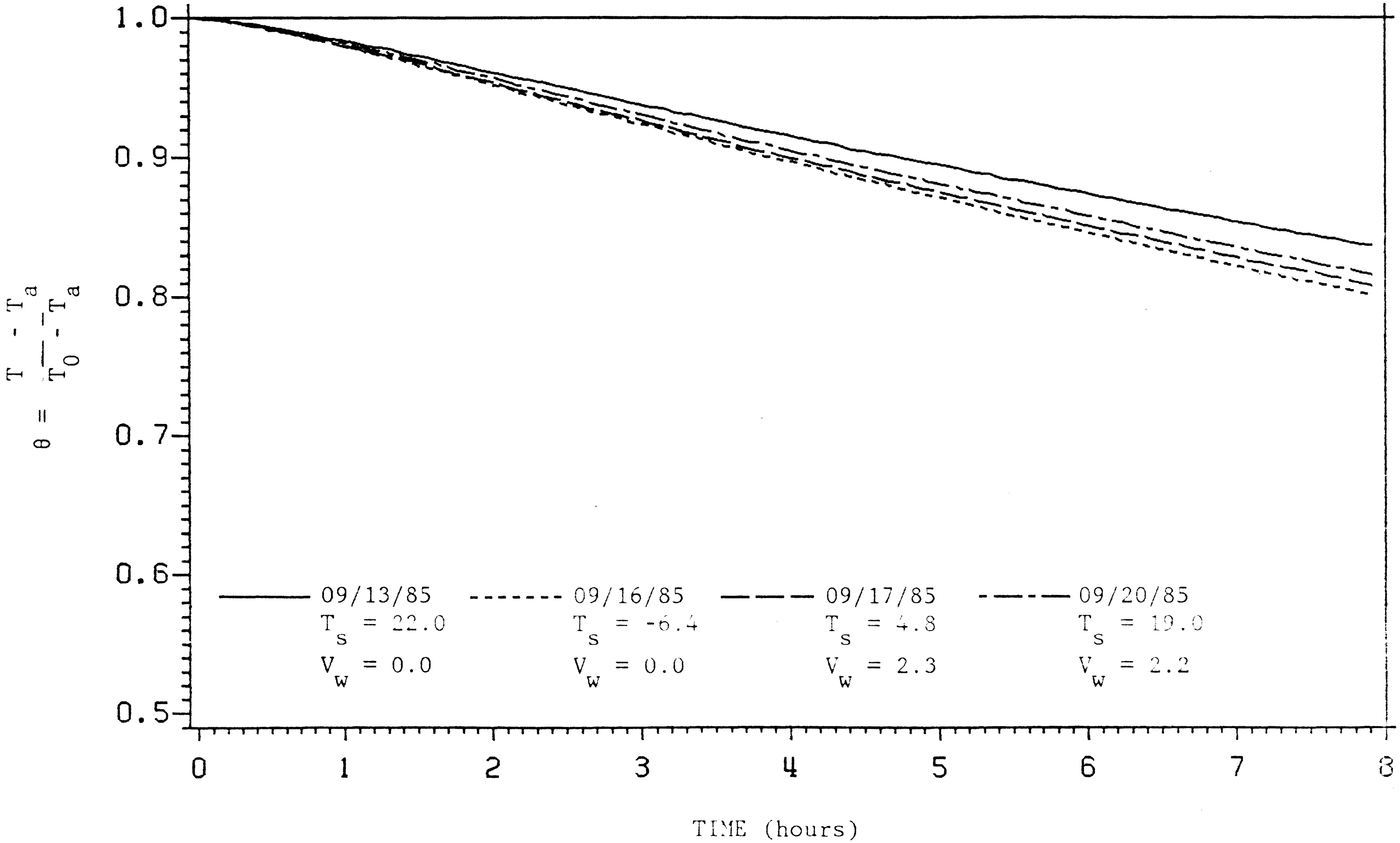

Figure 29. Comparison of Results from Indoor Heat Loss Tests 09/13/85, 09/16/85, 09/17/85, 09/20/85

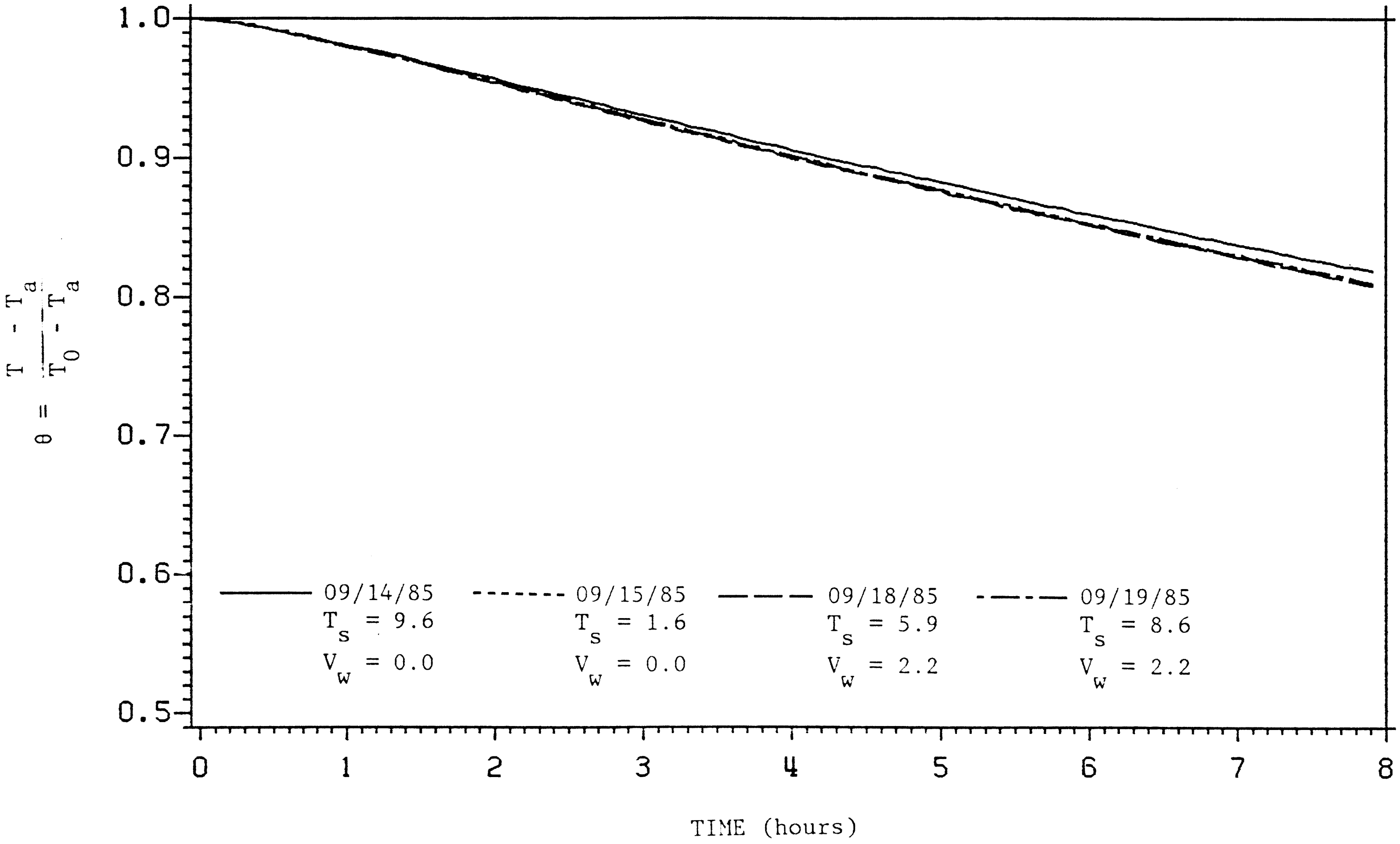

Figure 30. Comparison of Results from Indoor Heat Loss Tests 09/14/85, 09/15/85, 09/18/85, 09/19/85

studied for a medium wind speed of about 2.2 m/s. In this comparison, a change of 14 C in sky temperature, for a constant wind speed, results in only a 5 per cent change in heat loss. It is seen that with the introduction of wind, the effects of sky temperature are less pronounced and for changes in T_s of 4 C, the trends are almost completely obscured. Although it would be desirable to have tested lower sky temperatures at this wind speed, it was not possible because of the lack of cooling capacity of the chiller and frost buildup on the sky shield.

Figure 29 presents the results of a comparison of tests with varied sky temperatures and wind speeds. The test performed on 09/13/85, was for the case of no wind and a sky temperature of 22 C that was governed by the sky shield which was neither heated nor cooled but simply in equilibrium with the ambient air and ICS system underneath. This case produced the smallest value of heat loss. The 09/16/85 test with a sky temperature of -6 C and no wind only caused a 3.8 per cent increase in heat loss compared to the case with a sky temperature of 4.8 C and a wind speed of 2.3 m/s. However, the 09/17/85 test with a sky temperature of 4.8 C caused a 5 per cent increase in heat loss as compared to the 09/20/85 test with a sky temperature of 19.0 C at the same wind speed. The role that the sky temperature and wind speed play together in this case is obvious, but each separate effect is difficult to identify.

Figure 30 shows a comparison similar to that shown in Figure 28. The tests on 09/15/85, 09/18/85, and 09/19/85 resulted in exactly the same amount of heat loss. Therefore, when the sky temperatures are within ±4 C, a change in wind speed from 0 to 2.2 m/s completely obscures the differences in heat loss resulting from the differences in sky temperature.

In this case, the synergistic effect between the sky temperature and the wind speed is evident.

From the results presented in this section, it can be seen that both the sky temperature and wind speed play a significant role in heat losses from an ICS system. More specifically, it appears that the introduction of wind into the tests tends to overshadow small changes in sky temperature. Larger changes in sky temperature when included with wind, as seen in Figure 29, tend to be more visible in their effects on heat losses. When the sky temperature is reduced without any wind, the effects on the heat losses are significant. Therefore, it can be concluded that as a minimum, these two environmental parameters need to be measured and reported when conducting standardized certification or rating tests.

4.4 SRCC TYPE TEST RESULTS

Several tests were run indoors, similar to the indoor tests of the previous section, except they ran for a duration of 16 hours. The initial tank fluid temperature was again 60 C, the ambient temperature was set at 22 C (instead of 20 C as in the previous sections) and various cases were run for different sky temperatures and wind speeds. These tests are similar to the SRCC heat loss tests. The results are plotted in Figure 31 through Figure 34 and are tabulated in Appendix C.

In Figure 31, it is shown that for a constant sky temperature of 12.7 C, an increase in wind speed from 0 to 5.5 m/s caused a 5.9 per cent increase in heat loss. This is also verified in Figure 32, which for an almost constant sky temperature of about 30 C and a wind speed increase of 0 to 4.3 m/s resulted in an increase in heat loss of about 12.4 per cent. (While these sky temperatures are unrealistic for outdoor conditions, they might be realistic indoors under a solar irradiance simulator.)

Figure 33 and Figure 34 show the effect of sky temperature while maintaining a constant wind speed. For a high wind speed of approximately 5.6 m/s, a change in sky temperature of approximately 9 C results in a 3.7 per cent change in heat loss in Figure 33. Whereas in Figure 34, for a 21.7 C decrease in sky temperature and a wind speed of 0 m/s, a 15.7 per cent increase in heat loss results.

A summary for all the tests, from smallest heat loss to largest heat loss, is as follows:

- 10/28/85 $T_s = 30.9$ C $V_w = 0.0$ m/s $q_{loss} = 5.88$ MJ
- 10/25/85 $T_s = 22.6$ C $V_w = 0.0$ m/s $q_{loss} = 6.08$ MJ
- 11/14/85 $T_s = 12.7$ C $V_w = 0.0$ m/s $q_{loss} = 6.42$ MJ
- 10/22/85 $T_s = 21.4$ C $V_w = 5.6$ m/s $q_{loss} = 6.52$ MJ
- 10/31/85 $T_s = 28.8$ C $V_w = 4.3$ m/s $q_{loss} = 6.71$ MJ
- 10/23/85 $T_s = 12.2$ C $V_w = 5.7$ m/s $q_{loss} = 6.77$ MJ
- 11/11/85 $T_s = 12.7$ C $V_w = 5.5$ m/s $q_{loss} = 6.82$ MJ
- 10/24/85 $T_s = 0.8$ C $V_w = 0.0$ m/s $q_{loss} = 7.21$ MJ

Generally, the trends and observations for the 16 hour SRCC type tests are about the same as the 8 hour indoor tests of the previous section. The main difference is that an average ambient temperature of 22 C was used instead of 20 C, as 22 C is the required temperature specified by the associated rating organizations. Also, the magnitude of the change in the tank fluid temperature from the initial condition to the final condition is greater and the measurement uncertainty for this variable will have less of an effect on the overall results. Therefore, the heat losses can be measured more accurately for this 16-hour period.

Again, a difficulty encountered in testing was that it was impossible to study low sky temperatures with a high wind speed due to the lack of cooling capacity of the chiller. This situation, unfortunately, limited the range of tests that could be performed.

The initial and final tank fluid temperatures, average sky temperature, wind speed, and ambient temperature, heat loss, and the overall loss coefficient are summarized in Table 2.

4.5 FORCED CIRCULATION TESTS

The effect of continuously mixing the fluid in the tank of the ICS system on the heat losses from that system was studied. Several tests were run, both indoors and outdoors, and the results are tabulated in Appendix C and listed in Table 2.

After examining the experimental results for indoor forced tests 09/12/85, 09/21/85, and 11/03/85 and the outdoor forced tests 09/05/85 and 09/26/85, it can be seen that for all these cases the values of the lost energy and the overall loss coefficient are unexpectedly small.

Comparing the results between the indoor forced case 09/21/85 and the indoor stratified test of 09/15/85, the 09/21/85 case resulted in a smaller heat loss; the difference between the two was 33 per cent. This result is unrealistic considering the initial fluid temperatures are the

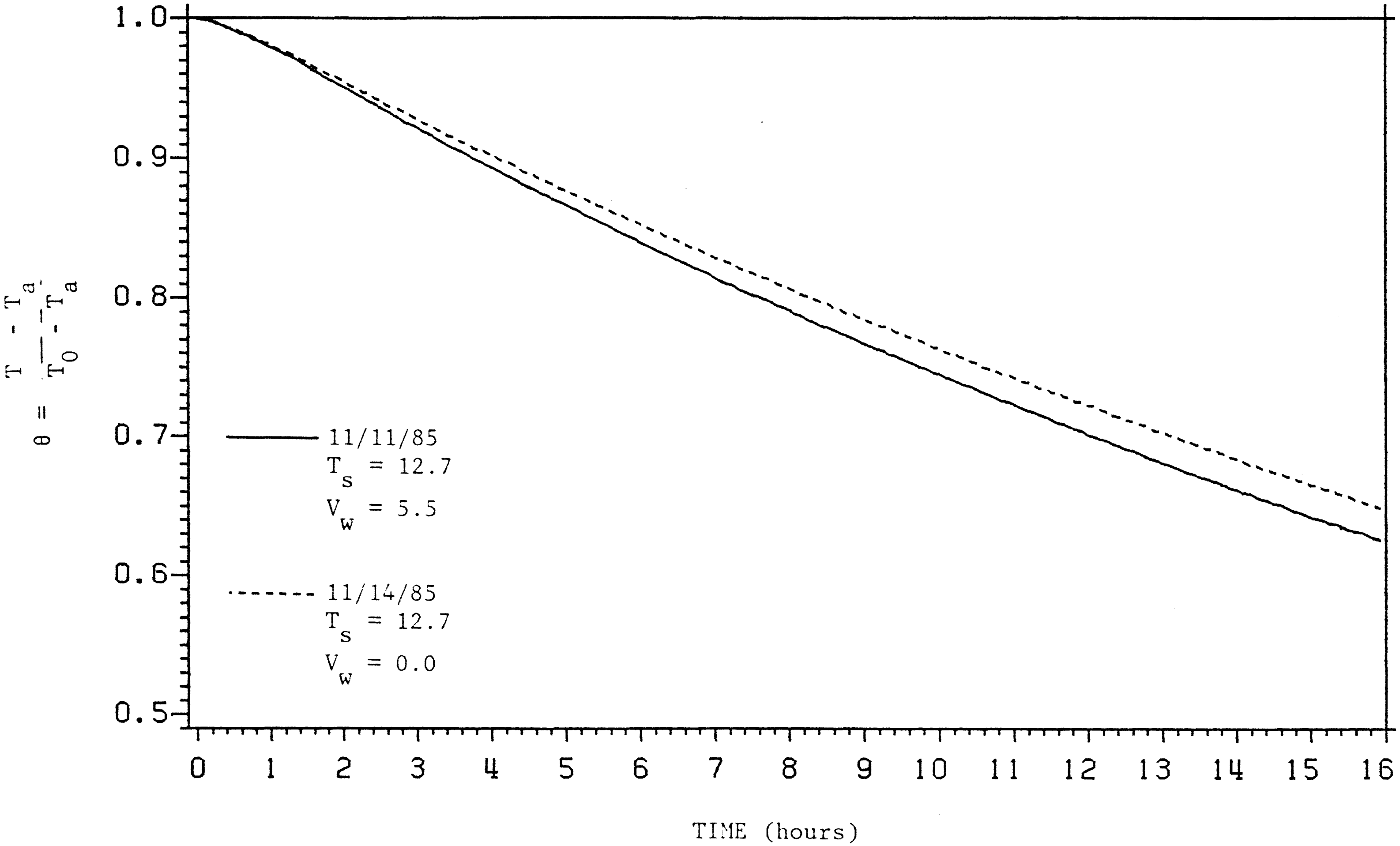

Figure 31. Comparison of Results from SRCC Type Heat Loss Tests 11/11/85 and 11/14/85

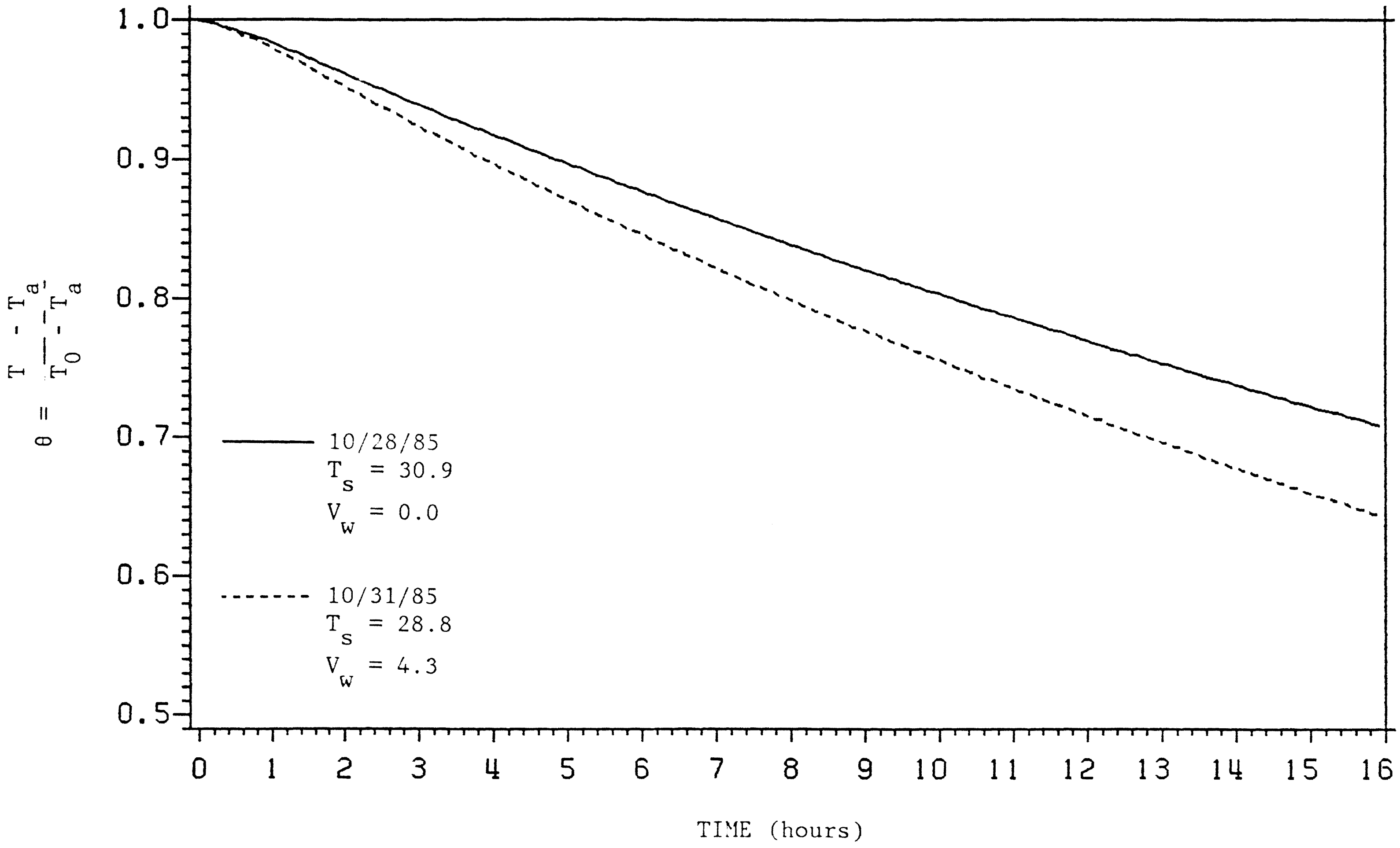

Figure 32. Comparison of Results from SRCC Type Heat Loss Tests 10/28/85 and 10/31/85

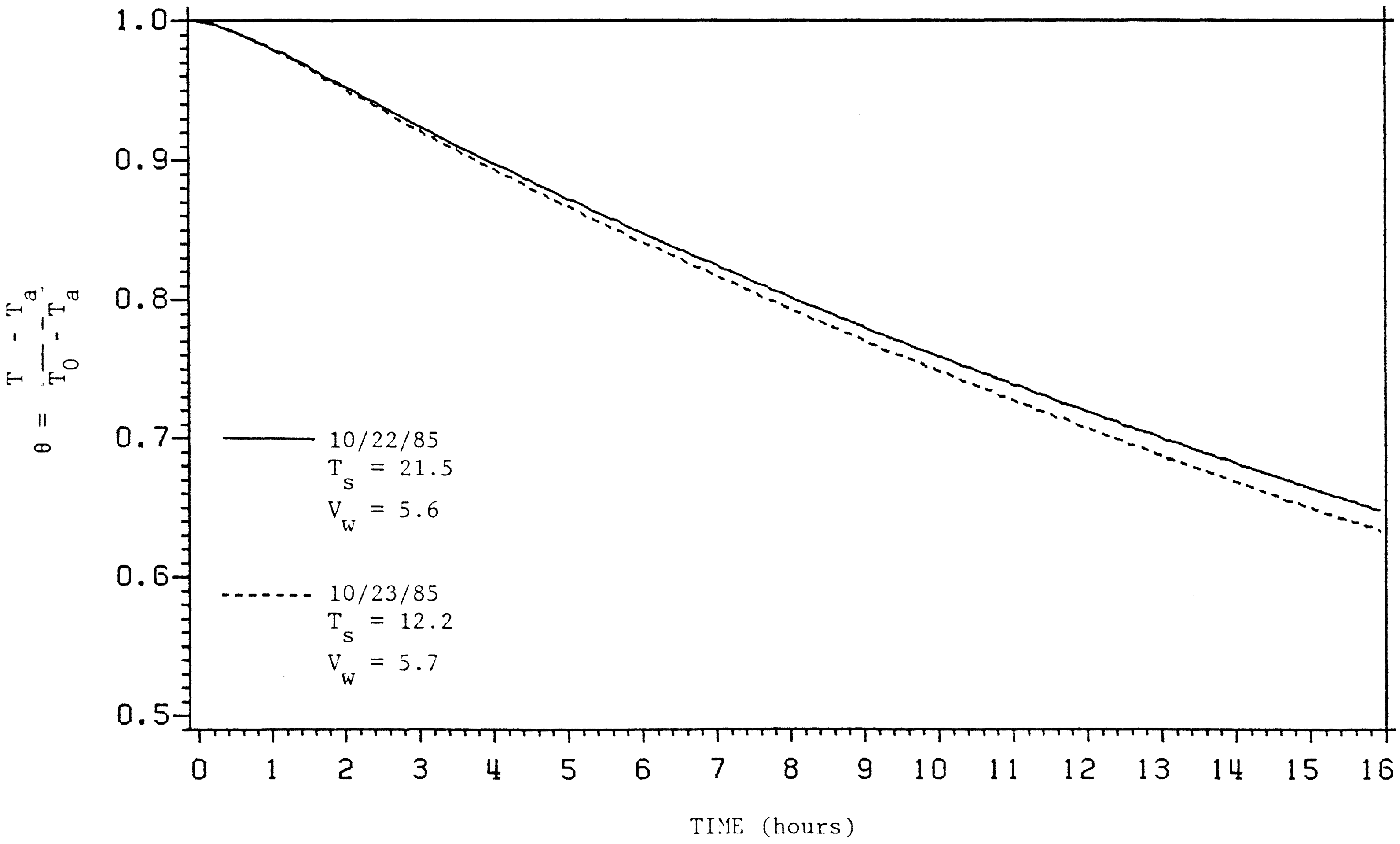

Figure 33. Comparison of Results from SRCC Type Heat Loss Tests 10/22/85 and 10/23/85

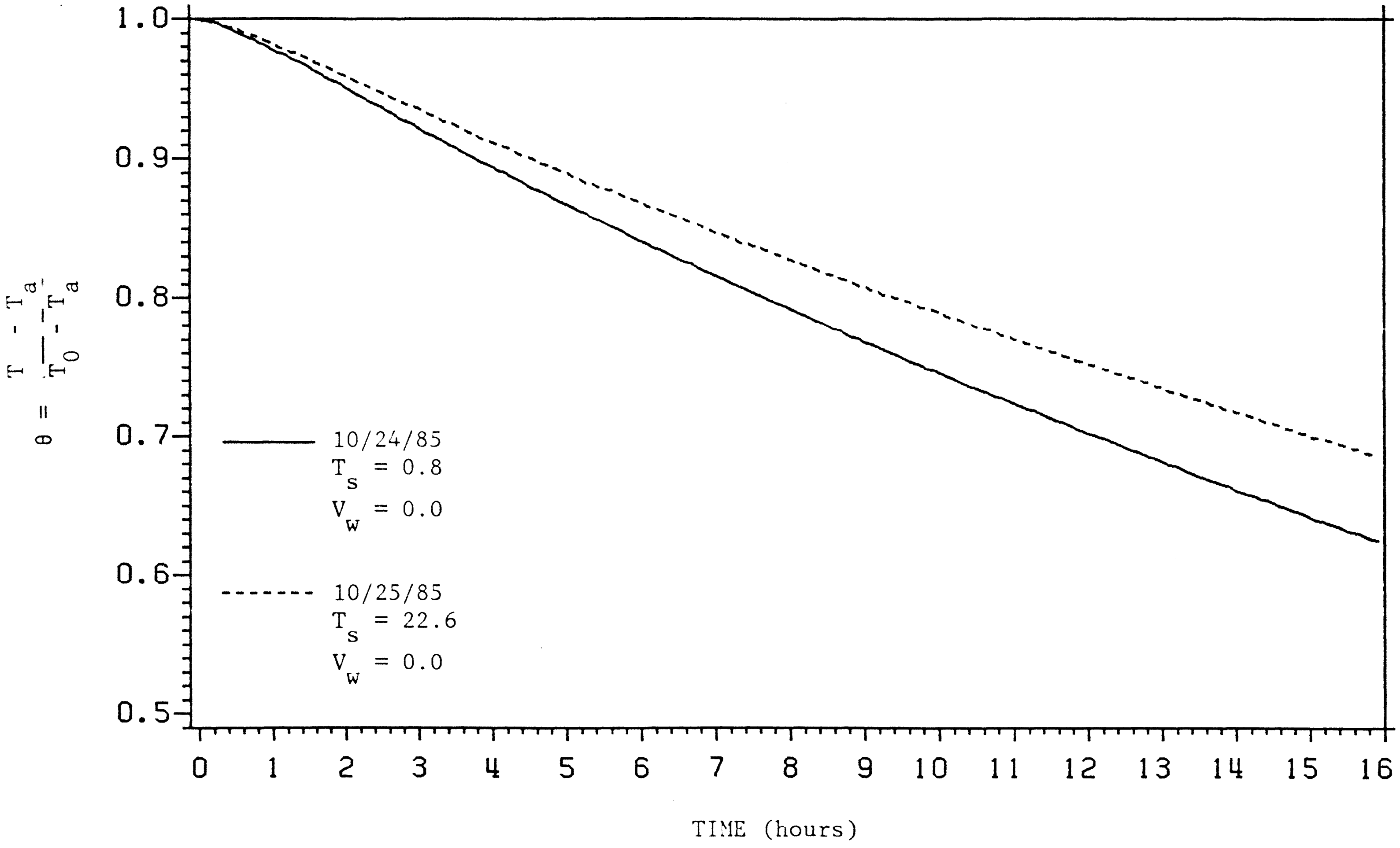

Figure 34. Comparison of Results from SRCC Type Heat Loss Tests 10/24/85 and 10/25/85

same and the sky temperature for the 09/15/85 case is 5 C higher. Also, for all of the stratified cases the overall loss coefficient is somewhere between 2 - 3 W/m^2 C. For all of the forced circulation cases the values are around 1.5 W/m^2 C and lower. Therefore, it can be concluded that there was significant heat addition to the fluid from the circulating pump. For this type of test, there can also be some losses through the external piping but it appears that the heat addition overrode those losses.

From the results of this section, it is evident that measuring the heat losses from an ICS system with the use of a circulating pump would not be an accurate alternative unless the heat addition from the pump could be taken into account. Also, circulating the fluid in the storage tank is not a natural operating mode for this class of collectors and is therefore not recommended. However, the mixing at the end of a test for 1 or 2 five-minute cycles results in a minimal amount of heat addition and can therefore be used with negligible error.

4.6 COMPUTER SIMULATION

4.6.1 VALIDATION OF FIRST-ORDER SYSTEM ASSUMPTION

A computer model developed by Lindsay [12] was used to simulate the nighttime losses from an ICS system. Since this collector is modeled as a first-order system, and this is generally an assumption that others have made as well, it was desired to see how well the actual data supported this assumption. Two outdoor tests and two SRCC type tests that were examined previously were chosen for comparison purposes. The results are plotted in Figure 35 through Figure 38 with the natural logarithm of the nondimensional temperature variable, θ, plotted against time. If the ICS system does behave as a first-order system, then the plots will yield a straight line.

After examining Figure 35 to Figure 38 it appears that all of the cases are almost perfectly linear. However, the results from the two SRCC type indoor tests, 10/31/85 and 10/28/85 are in slightly better agreement with the first-order assumption. This is probably due to the fact that they are under constant environmental conditions throughout the entire test period, whereas the environmental parameters fluctuate in the outdoor tests. Therefore, it can be concluded that modeling this collector as a first-order system is a good assumption.

4.6.2 DISCUSSION OF RESULTS

The computer model for the ICS system used in this investigation was run for several different indoor and outdoor cases. Unfortunately, the model does not predict the tank temperatures well; it underpredicts the temperature for the first couple of hours and overpredicts the temperature for the last couple of hours. When examining the ratio of the heat loss for the simulated case to the measured case, the model consistently gave results 35 - 45 per cent less than measured. In other words, the model always underpredicts the heat losses.

Adjustments were made to determine the shortcomings of the program but attempts were unsuccessful. If a sky temperature much less than the

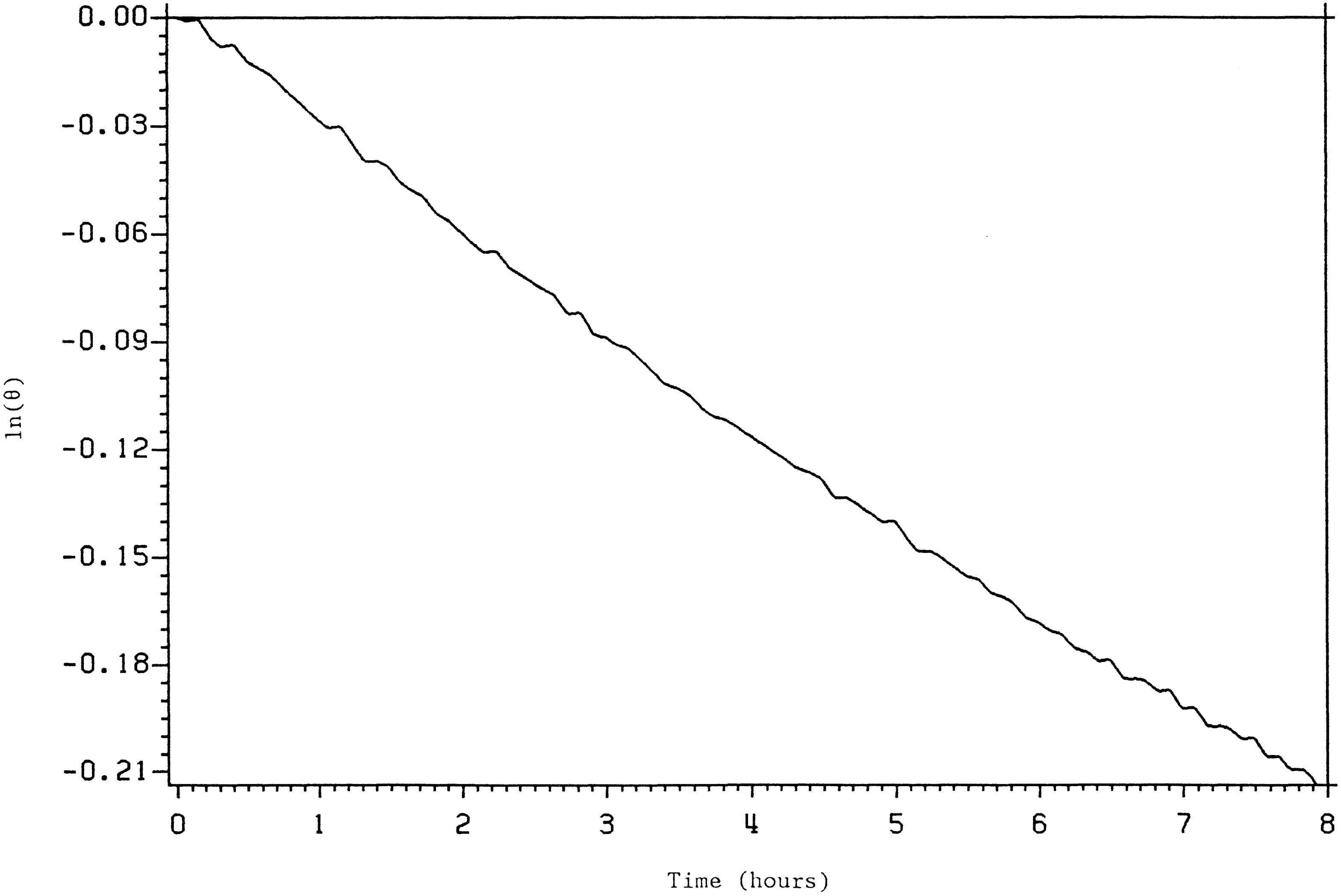

Figure 35. Results from Outdoor Heat Loss Test 09/25/85 for Determination of the Validity of a First Order System

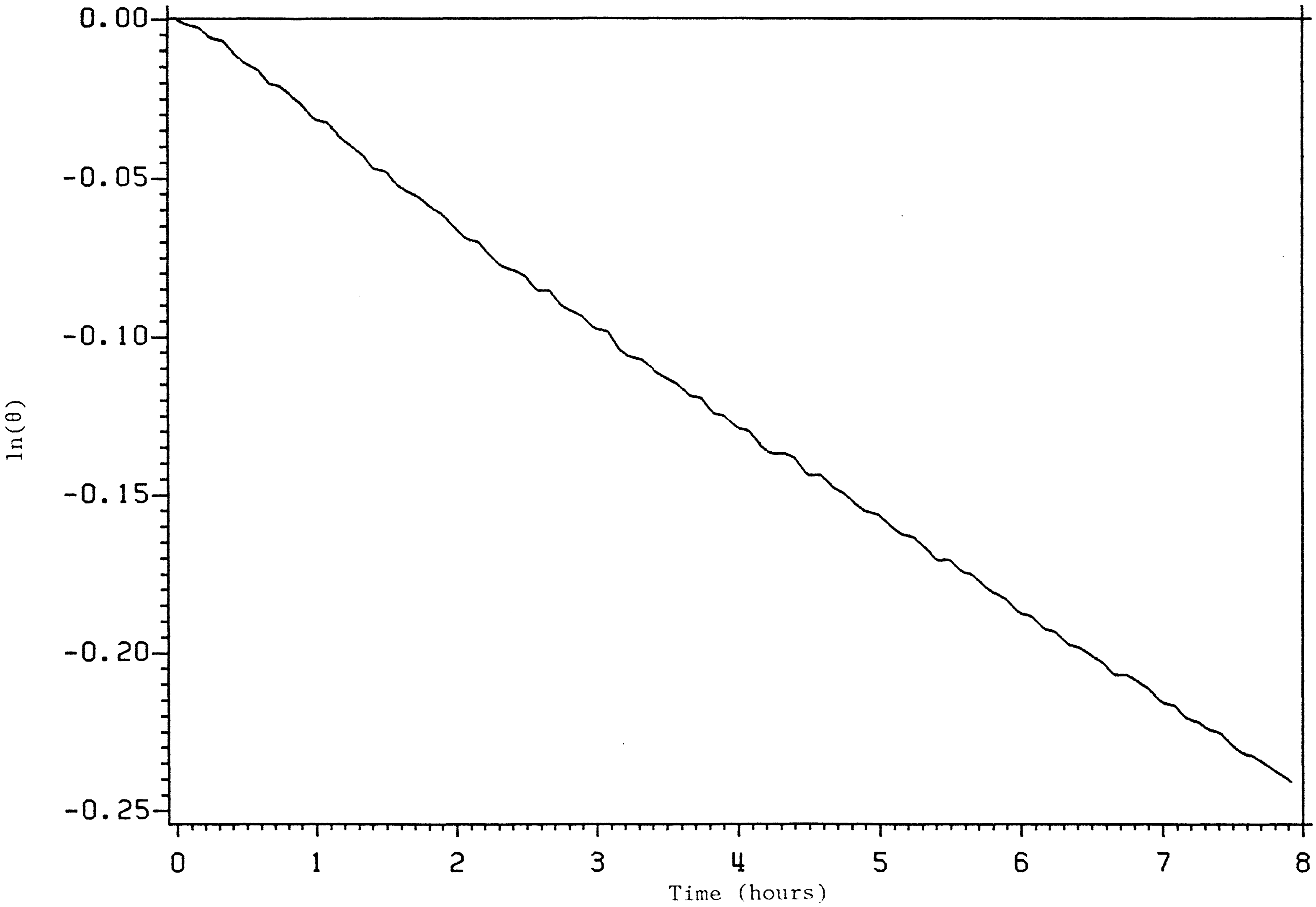

Figure 36. Results from Outdoor Heat Loss Test 10/10/85 for Determination of the Validity of a First Order System

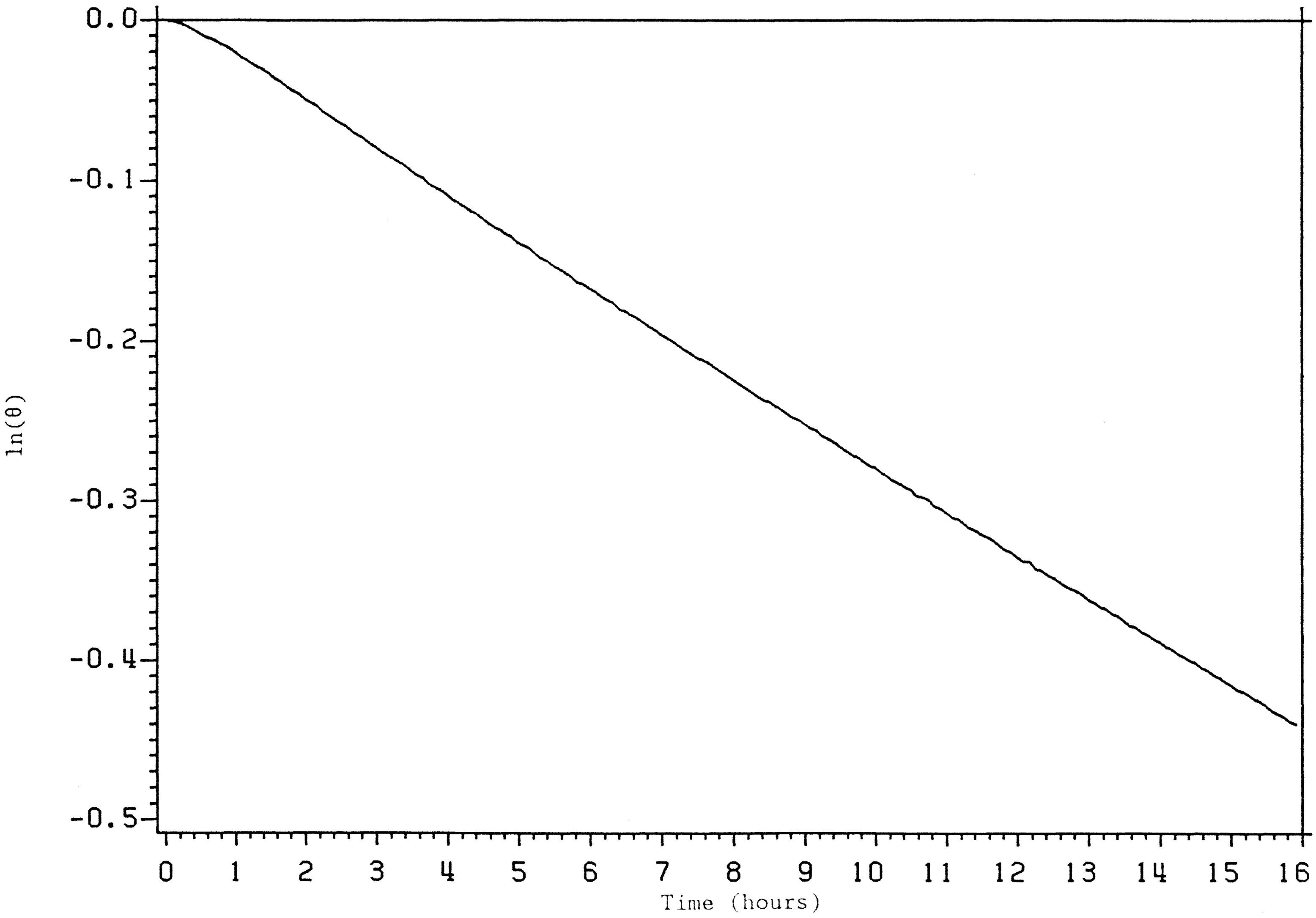

Figure 37. Results from SRCC Type Heat Loss Test 10/31/85 for Determination of the Validity of a First Order System

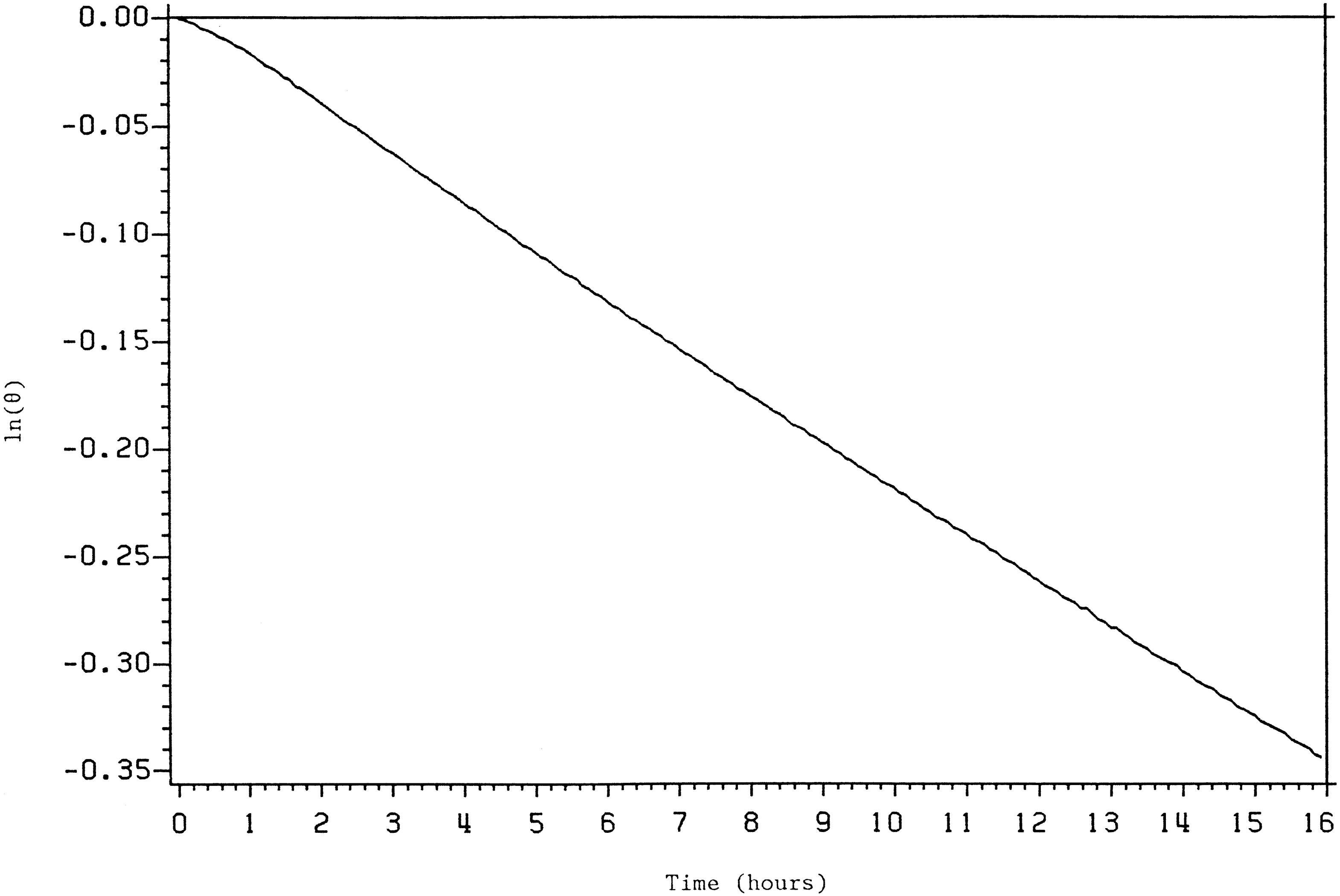

Figure 38. Results from SRCC Type Heat Loss Test 10/28/85 for Determination of the Validity of a First Order System

actual sky temperature (anywhere from 40 C to 15 C below the actual value) was entered, a reduction in percentage points of 3 per cent to 16 per cent could be realized. However, the error is still too high to use the model with any confidence. Because of the complexity of this model, and its relative shortcomings, it was not used further in this investigation.

4.7 DISCUSSION

The results of the previous sections brought up many interesting and unexpected findings. The most important point now is to determine how the current rating standards can be improved to yield more accurate results.

Currently all ICS systems that are tested for a standard rating are required to undergo a heat loss test. This test is performed indoors and lasts for 16 hours. As seen in the previous sections, these requirements are important, as outdoor tests are difficult to standardize due to weather conditions and errors associated with measurement uncertainty will be much less over a 16-hour period. The standard test requires a wind speed of 3.4 m/s (±0.8 m/s) across the collector and it sets no tolerances on the sky temperature. From the results presented in this chapter, it can be seen that both the sky temperature and wind speed significantly affect the heat losses from this type of collector.

As can be seen from the SRCC type 16-hour test results of Section 4.4, an increase in wind speed from 0 to 5.5 m/s caused a 5.9 per cent increase in heat loss for a sky temperature of 12.7 C, and an increase in wind speed from 0 to 4.3 m/s resulted in a 12.4 per cent increase in heat loss for a sky temperature near 30 C. If it is assumed that the sky temperature during standard indoor heat loss tests is equal to ambient temperature, and if the average of the previous results from the SRCC type tests is used, then for a sky temperature of 21 C, an increase in wind speed from 0 to 5 m/s results in a 9 per cent increase in heat loss. Now, for a wind speed of 5.6 m/s, a 3.7 per cent increase in heat loss was found for a 9 C decrease in sky temperature. If that sky temperature were further reduced to about 15 or 20 C below ambient, then the reduction may have approximately the same effect on heat loss as the change in wind speed. This approximation is made because, as mentioned previously, very low sky temperatures were not available at high wind speeds. The point is, for a large enough change in sky temperature, the effects on heat loss are the same as for a change in wind speed. However, a prescribed wind speed is used in heat loss testing for ICS systems (at a value of 3.4 m/s which would induce less than a 9 per cent increase in heat loss for a T_{sky} of 21 C) and the sky temperature is not.

The outdoor data taken over a couple of months showed very few cases when the wind speed got over 1 m/s. However, in every case the sky temperature is always more than 5 C below ambient temperature. Therefore, it appears that a decrease in sky temperature may affect an ICS system on almost all nights of operation, whereas at this particular site, the effect of wind may be realized on a much smaller fraction of nights.

Since ICS systems are highly susceptible to nighttime losses, many manufacturers have incorporated heat retention features into their design. It is important that this class of collectors be tested under a sky temperature range where the advantages of this design attribute would be realized in the test results and performance ratings.

An important point that should be noted is that tests were only performed on one type of ICS system in this investigation. This particular collector is well designed to minimize the convective and infrared radiative losses. Therefore, the heat losses may tend to be less than would be realized for another system.

A recommendation to improve the credibility of the heat loss test results is to test the ICS system under an artificial nighttime sky, which is held at a constant temperature in the range between 0 - 5 C, while maintaining an ambient temperature of 20 - 22 C, and a constant wind speed of 3.4 m/s. The sky temperature plays an equally important role as wind speed in the real operating life of many ICS systems. Therefore, tests should be conducted under natural conditions. As a minimum requirement, the sky temperature should be measured with a pyrgeometer and reported.

5.0 CONCLUSIONS

An experimental investigation was conducted to study the nighttime losses from one type of ICS solar domestic hot water system. This system was well designed to minimize both convective and radiative losses. The following conclusions resulted:

1. The energy stored in the nonfluid elements of the collector is negligible. The heat stored within the fluid can be calculated accurately by taking the average of two thermocouples, located in the inlet and discharge connections of the collector, for the case of well mixed fluid.

2. Outdoor test conditions can be simulated indoors, providing the ambient temperature, sky temperature, and wind speed are adequately duplicated. The value of the heat loss was found to be within 8 per cent. Indoor tests resulted in lower heat loss for the same combination of environmental parameters.

3. Both wind speed and sky temperature affect the losses from an ICS system. The sky temperature significantly effects the heat loss for no wind and shows much less of an effect when wind is introduced. The wind speed tends to overshadow small changes in sky temperature.

4. The effects of ambient air temperature outdoors, over a range of 4 C to 21 C, were difficult to single out from the other environmental parameters and consequently, they were not dealt with separately.

5. The assumption that ICS systems act as a first-order system is valid. An existing mathematical model available to predict performance of the ICS system always underpredicted the tank temperatures and heat losses, and consequently was not used for the purpose of determining the heat losses.

6. The indoor test procedure that is currently used should be changed to include testing under an artificial nighttime sky with a sky temperature of 0 C - 5 C, (with the existing requirement of an ambient temperature of 22 C and a wind speed of 3.4 m/s), to yield more credible results.

7. A minimum requirement that should be included in ICS heat loss tests is to monitor and report the sky temperature for the duration of the test period.

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APPENDIX A. INSTRUMENTATION

Calibration: All of the instruments listed in Table 3, with the exception of the windvane, were calibrated prior to testing. The pyrgeometer was sent to the Eppley Laboratory for calibration and a constant of 4.67E-6 $v/(W/m^2)$ was determined. This constant is based on the assumption of a linear relationship between radiation intensity and emf.

Both of the 3-cup anemometers were calibrated in a subsonic wind tunnel. Values were recorded at several points up to the maximum wind speed and then were checked for repeatability by taking several points while decreasing the wind speed. Figure 39 shows the data points and the resulting least squares fit from the calibration on the anemometer used indoors. A constant of 42.4 (m/s)/v was determined with a maximum deviation of about 0.4 m/s. Figure 40 shows the data points and the resulting least squares fit from the calibration on the anemometer used outdoors. A constant of 16.2 (m/s)/v was determined with a maximum deviation of about 0.5 m/s.

The thermocouples used indoors were calibrated with the HP3497A data acquisition system and the HP9236 microcomputer. The calibration was for the system because the least count of the data acquisition equipment was taken into account as well. Voltages were read from the thermocouples into the HP3497A and then converted into temperatures with software on the HP9236. Values were then compared to temperatures read off ASTM thermometers. It was found that the thermocouples read within ±0.2 C of the thermometer, which was well within the manufacturers specification

Table 3. Instrument Manufacturers Listing

Instrument or Recording Device	Manufacturer	Model Number
Infrared Radiometer (Pyrgeometer)	Eppley Laboratory, Inc.	PIR
3-cup Anemometer (outdoors)	Weathermeasure	W103-B
3-cup Anemometer (indoors)	Weathermeasure Weathertronics	2011
Lightweight Vane	Weathermeasure	W104-2
Pump	Grundfos	UPS 20-42
Thermocouples	Custom Built	20 Gage Type T
Electronic Integrators	AGM Electronics, Inc.	TA 4013 W/ 10 TA 4028 & 10 TA 4011
Data Acquisition/ Control Unit	Hewlett Packard	HP 3497A

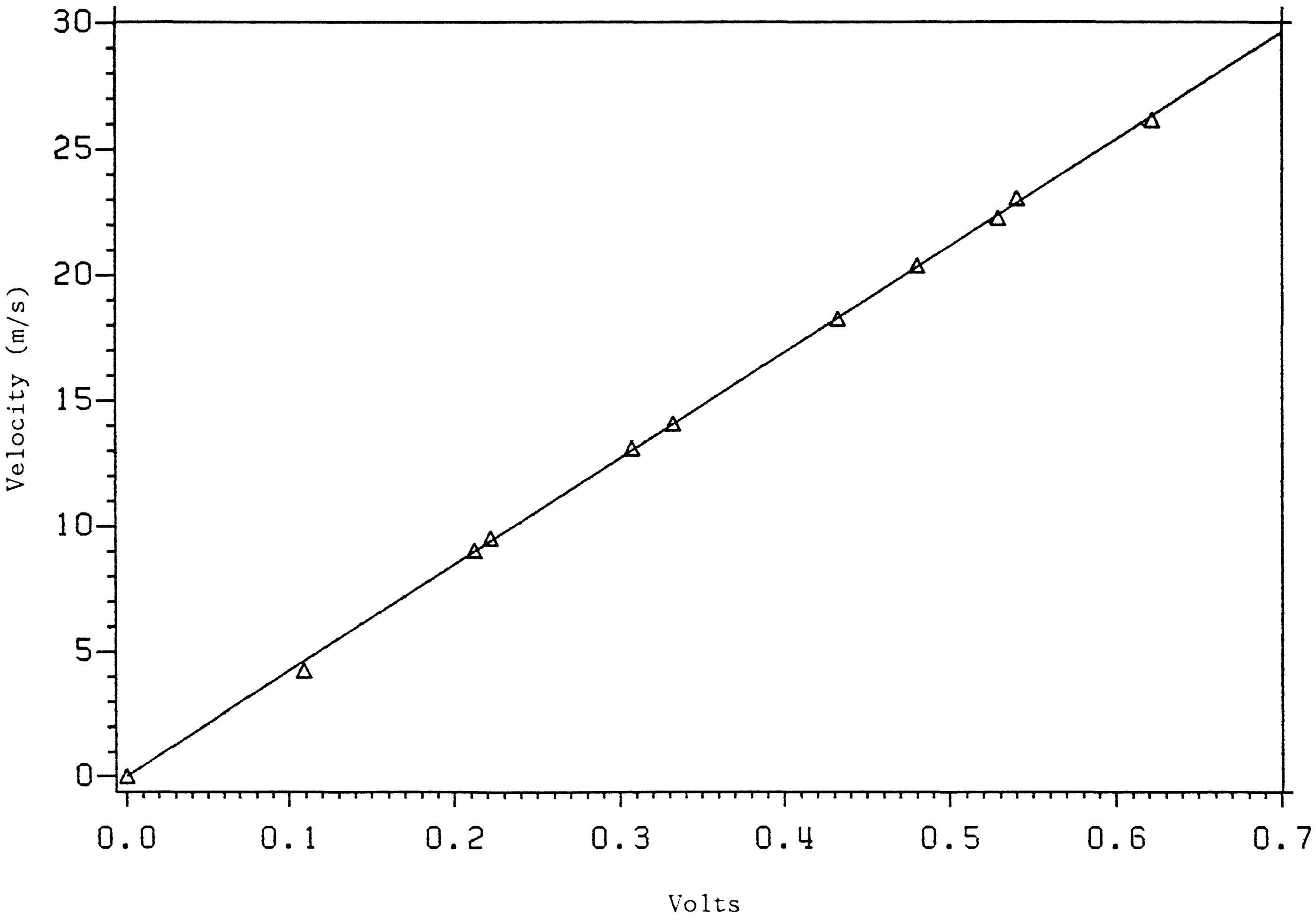

Figure 39. Calibration Curve of Weathermeasure Weathertronics Model 2011 Anemometer

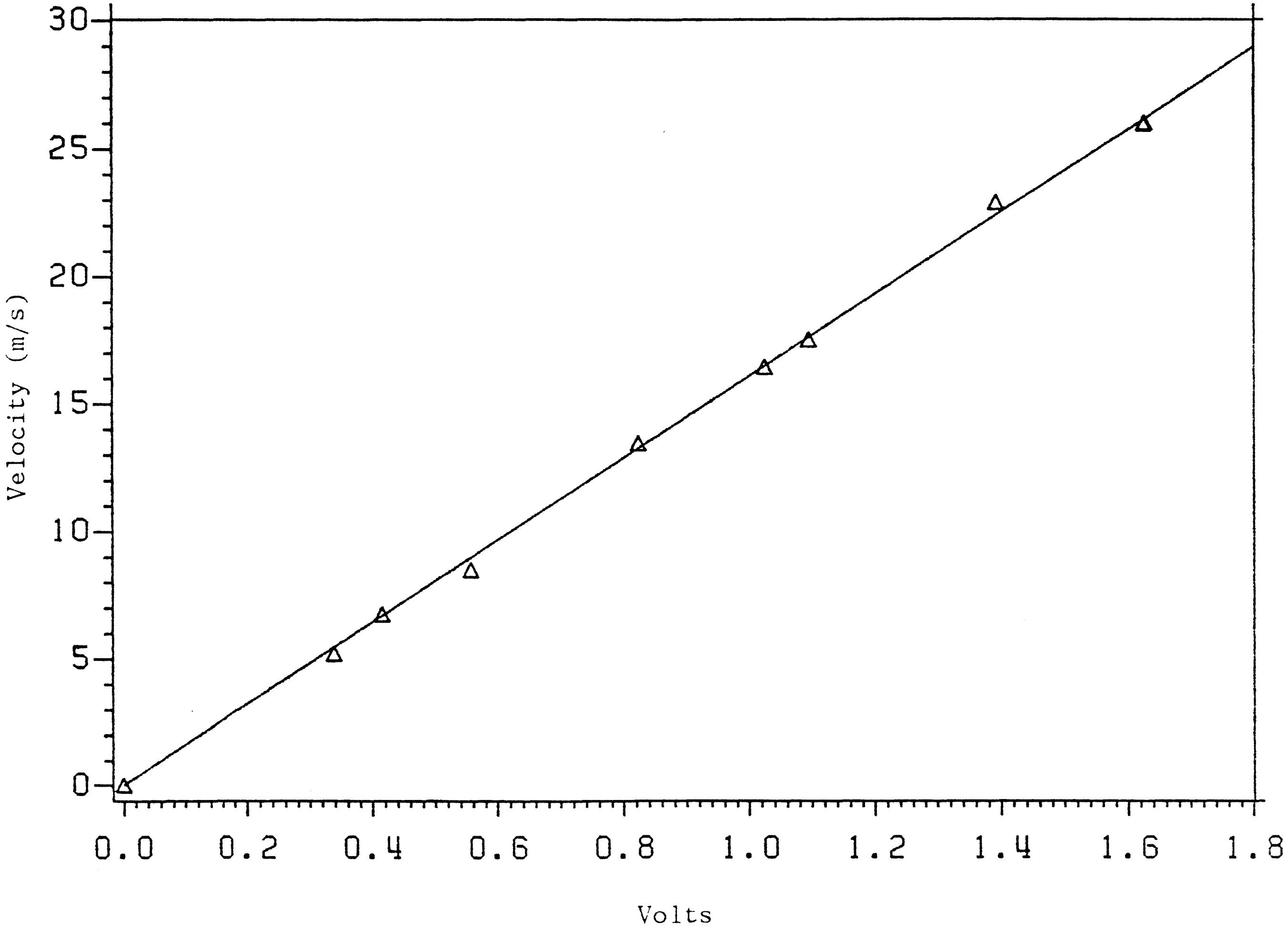

Figure 40. Calibration Curve of Weathermeasure W103B Anemometer

of ±0.5 C. Therefore, no calibration curves were used and the voltages were converted directly.

The thermocouples used outdoors were calibrated with the AGM Electronic Integrators and the TRS-80 Model III microcomputer. The calibration was again for entire the system. Several readings were taken at several temperatures and compared with ASTM thermometers. There was a discrepancy between the measured temperature and the actual temperature and a calibration curve was established as shown in Figure 41. The difference between the actual temperature and the temperature calculated by the system, $\Delta = T_{act} - T_{meas}$, is plotted against the temperature calculated by the system, T_{meas}. The calibrated temperatures, T_{top} and T_{bot}, can then be found from each respective equation of the least squares curve, as shown in Figure 41. T_{top} is calculated as

$$T_{top} = T_{meas_t} + T_{meas_t}(-0.00441) + 0.15 \tag{14}$$

and T_{bot} is calculated as

$$T_{bot} = T_{meas_b} + T_{meas_b}(-0.00562) + 0.60 \tag{15}$$

These equations, (14) and (15), yield the calibrated values of the tank temperatures which were used whenever data was taken. T_{top} is within ±0.11 C of the actual temperature and T_{bot} is within ±0.07 C of the actual temperature.

The thermocouple used to measure ambient air temperature was well within the accuracy limits required to read that variable.

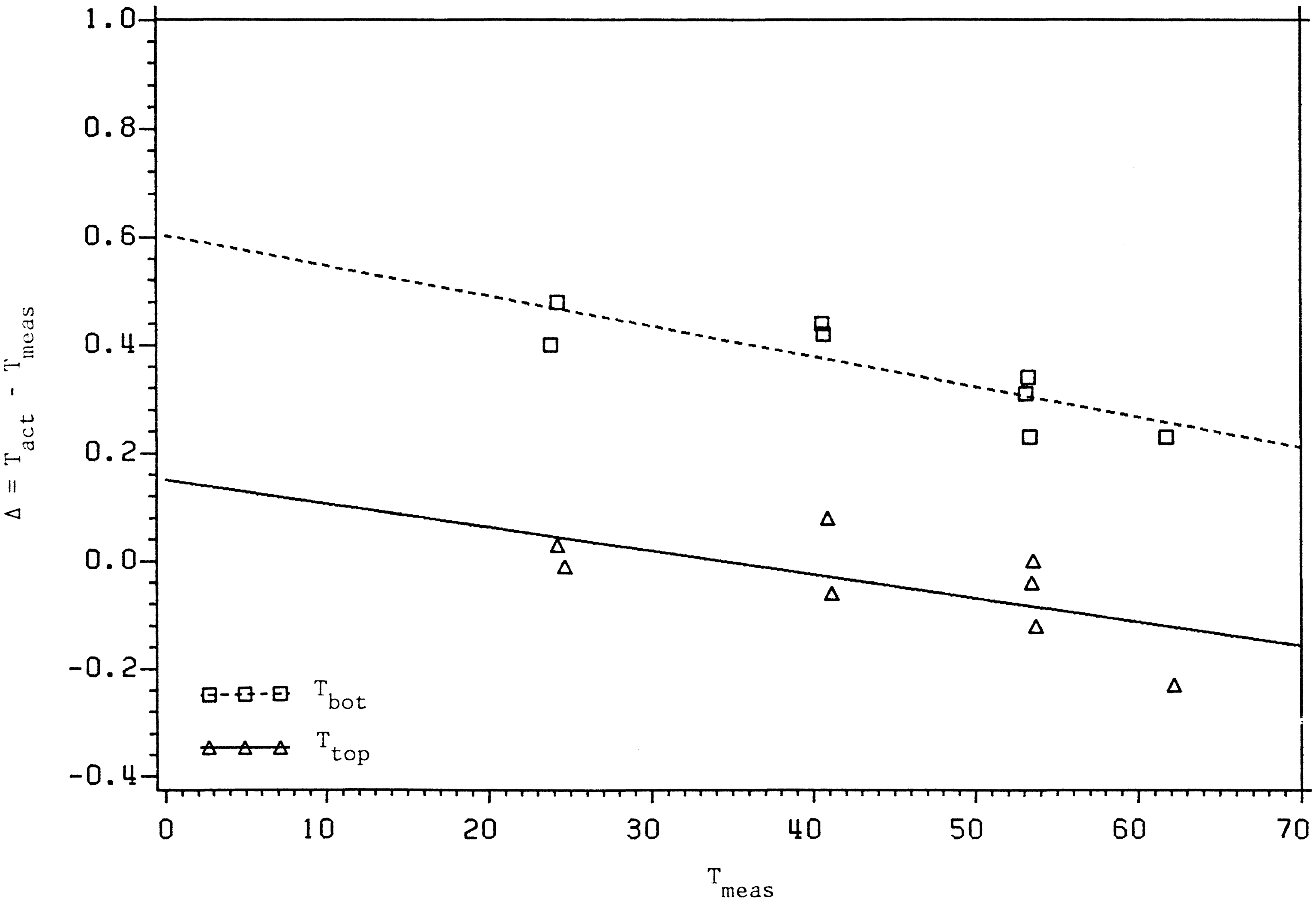

Figure 41. Calibration Curve of Thermocouples used Outdoors

APPENDIX B. ANALYSIS

The losses from the absorber-tank of the ICS system are governed by thermal radiation and convection between the various collector components and the air within the collector enclosure. In order to solve for the overall loss coefficient an iterative solution procedure is utilized.

The first step of the solution procedure is to determine the radiation configuration factors and the surface properties of the components. Hourly average values of the tank temperature, wind speed, sky temperature, and ambient temperature are read in for each of the eight hours of the test period that is being simulated. The initial tank temperature is then set based on the initial fluid temperature from the experimental data. The component temperatures are then estimated based on this mean temperature and the overall heat transfer coefficient is set equal to zero. A loop is started which uses these estimated temperatures and the initial loss coefficient to converge on actual values for the overall loss coefficient, component temperatures and radiosities. First, the radiosities are calculated, then the convection coefficients are determined and they are both used to solve the energy balances to yield updated component temperatures. A new value of the overall loss coefficient is determined and the process is repeated until the component temperatures and the overall loss coefficient converge.

With this new value of the overall loss coefficient the mean tank temperature is revised, the component temperatures are revised, and the entire process is repeated until the tank temperature converges. Once the mean tank temperature is determined for one hour the iteration procedure starts from the beginning to calculate values for the next hour.

At the end of the modeling process, the lost energy is calculated and the final average value of the overall loss coefficient is determined.

The details for calculating the shape factors, the radiosity distributions, and the convection coefficients, as well as the energy balances are presented in the next few pages. For further details of this model and the calculation procedures, the reader is referred to reference 12.

Radiation Configuration Factors: The radiation configuration factors were solved for using the summation rule and reciprocity relations for enclosures for all the surfaces except the absorber-tank to the cover. This was derived from a special relation for a cylinder to a rectangle as tabulated in Sparrow and Cess [40]. The shape factor from one cover surface to another was taken to be 1.

Infrared Radiation Analysis: In this investigation, the longwave radiation distribution is evaluated using a radiosity approach. The collector components are assumed to be isothermal with diffuse surface behavior, and the associated radiosities are shown in Figure 42. The equations which result from this analysis are listed below.

Absorber-Tank

$$J_t = \varepsilon_t \sigma T_t^4 + \rho_t (F_{tr} J_r + F_{tl} J_c)$$

Reflector

$$J_r = \varepsilon_r \sigma T_r^4 + \rho_r (F_{rt} J_t + F_{r1} J_c + F_{rr} J_r)$$

Covers

$$J_c = \varepsilon_1 \sigma T_1^4 + \rho_1 (F_{1t} J_t + F_{1r} J_r) + \tau_1 J_{c2}$$

$$J_{c1} = \varepsilon_1 \sigma T_1^4 + \rho_1 J_{c2} + \tau_1 (F_{1t} J_t + F_{1r} J_r)$$

$$J_{c2} = \varepsilon_2 \sigma T_2^4 + \rho_2 J_{c1} + \tau_2 J_{c4}$$

$$J_{c3} = \varepsilon_2 \sigma T_2^4 + \rho_2 J_{c4} + \tau_2 J_{c1}$$

$$J_{c4} = \varepsilon_3 \sigma T_3^4 + \rho_3 J_{c3} + \tau_3 \sigma T_s^4$$

$$J_s = \varepsilon_3 \sigma T_3^4 + \rho_3 \sigma T_s^4 + \tau_3 J_{c3}$$

These eight equations can be solved for the eight unknowns (J_t, J_r, J_c, J_{c1}, J_{c2}, J_{c3}, J_{c4}, J_s) by an appropriate mathematical technique. These radiosities can then be substituted into the energy balances shown in a later section.

Convection Coefficients: In order to calculate the convective heat transfer between components within the collector, two different heat transfer coefficients, h, must be determined. The first case is that between the parallel plates of glazing which make up the cover system. An experimental study by Hollands was used which presented a relationship between the Nusselt number and the Rayleigh number for tilt angles, β, from 0 to 75 degrees as

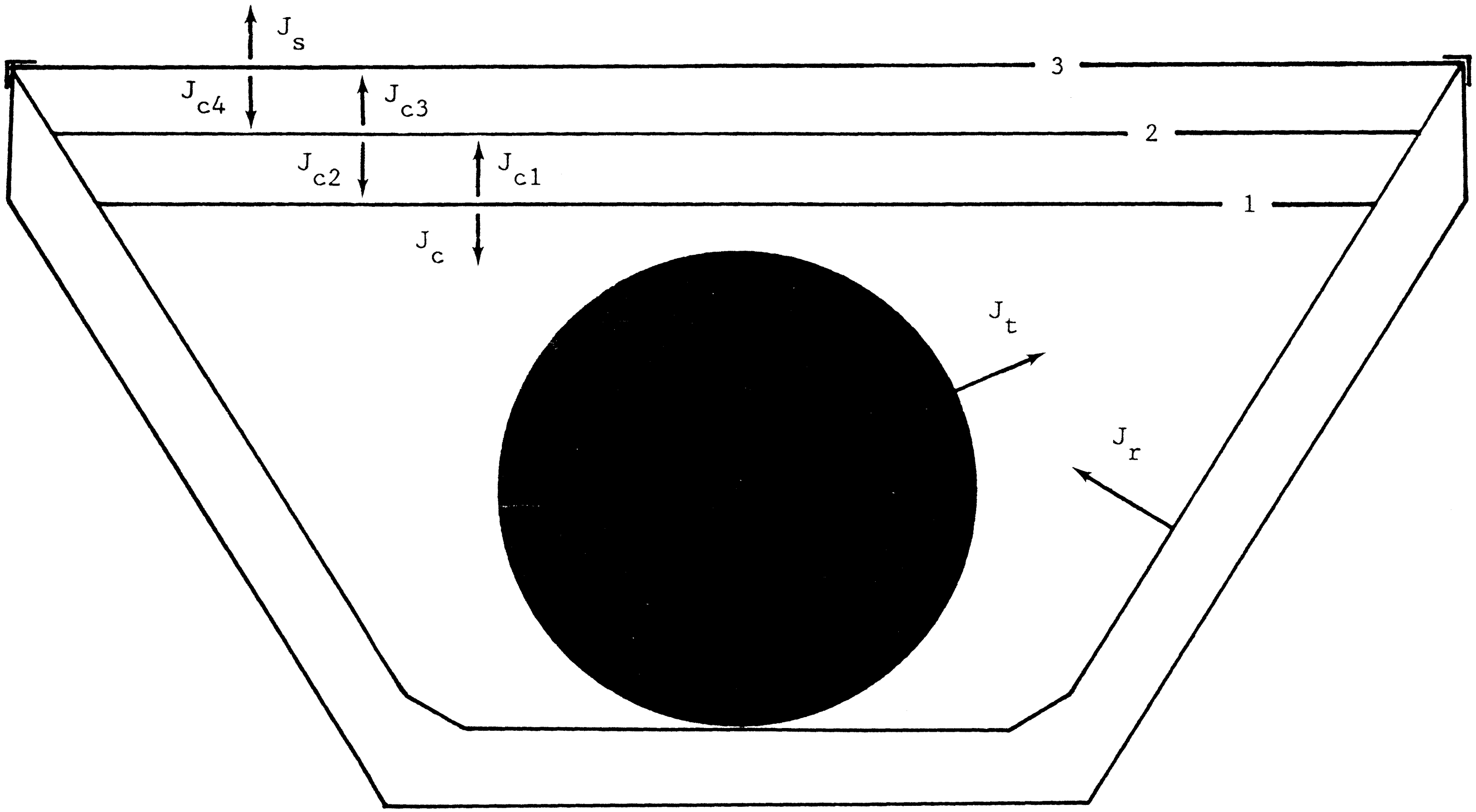

Figure 42. Longwave Radiosity Components (not to scale)

$$Nu = 1 + 1.44 \left[1 - \frac{1708}{Ra \cos \beta}\right]^{+} \left(1 - \frac{1708 (\sin 1.8\beta)^{1.6}}{Ra \cos \beta}\right) + \left[\left(\frac{Ra \cos \beta}{5830}\right)^{1/3} - 1\right]^{+} \qquad (16)$$

where the + exponent implies that only the positive values of the term in the square brackets are to be used [34]. The convection coefficient can then be determined by

$$h = \frac{Nu\ k}{d} \qquad (17)$$

where k is the thermal conductivity of the plate and d is the plate spacing.

The second case involves free convection between the absorber-tank and the various flat surfaces at different angles (i.e. the reflector sides, cover 1). A correlation of the Nusselt number used by Lindsay [12] was

$$Nu = 0.55\ (Gr\ Pr)^{0.25} \qquad (18)$$

as an approximate average of the horizontal cylinder and the horizontal flat plate cases. It should be noted that determination of the heat transfer coefficient is a very complex matter and that great simplifications were made to obtain this correlation.

The correlation for the wind-induced heat transfer coefficient that was used in this investigation is

$$h_w = 2.8 + 3.0\ V_w \tag{19}$$

The uncertainty in determining this convection coefficient is considerable and other authors suggest possibly better correlations but they were not used here. The reason is that some of the other correlations introduce another parameter that may increase the uncertainty. Therefore, for this application is was decided that the relation in equation (19) would be the most accurate.

Component Energy Balances: When evaluating the temperature of each component (i.e., the cover system, the internal reflector, the air within the enclosure) it is necessary to perform an energy balance about the surface of each component. It is assumed that there is no stored energy within these elements and that they are in thermal equilibrium with the storage tank and ambient air at all times. The heat transfer is assumed to be one-dimensional through these components. The resulting energy balances are listed below.

Reflector

$$\frac{\varepsilon_r}{(1 - \varepsilon_r)} (\sigma T_r^4 - J_r) - h_{rg}(T_g - T_r) + U_{ra}(T_r - T_a) = 0$$

Air in the Collector Enclosure

$$h_{tg}(T_t - T_g) - h_{lg}(T_g - T_c) - h_{rg}(T_g - T_r) = 0$$

Cover 1 (inner)

$$-\varepsilon_1(2\sigma T_1^4 - F_{1t}J_t - F_{1r}J_r - J_{c2}) + h_{1g}(T_g - T_1) - h_{12}(T_1 - T_2) = 0$$

Cover 2 (middle)

$$-\varepsilon_2(2\sigma T_2^4 - J_{c4} - J_{c1}) + h_{12}(T_1 - T_2) - h_{23}(T_2 - T_3) = 0$$

Cover 3 (outer)

$$-\varepsilon_3(2\sigma T_3^4 - \sigma T_s^4 - J_{c3}) + h_{23}(T_2 - T_3) - h_w(T_3 - T_a) = 0$$

It may be noted that these equations can be solved for the five unknown temperatures, with latest values of the radiosities and the convection coefficients. However, these quantities are also temperature dependent and an iterative procedure is necessary for the solution of these equations.

APPENDIX C. TABULATED SUMMARY OF TEST RESULTS

OUTDOOR FORCED TEST, 09/05/85

TEST STARTING AT 21:40

TEST DURATION = 8 HOURS

INITIAL TEMP = 53.5 C

FINAL TEMP = 52.0 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	53.42	53.38	0.48	11.19	22.03
2	53.24	53.17	0.01	10.62	21.15
3	53.06	52.98	0.09	9.75	20.46
4	52.89	52.82	0.14	9.25	19.99
5	52.70	52.64	0.27	9.02	19.44
6	52.52	52.44	0.00	8.28	18.85
7	52.32	52.24	0.00	8.13	18.57
8	52.10	52.01	0.44	7.70	18.30

LOST ENERGY = 0.736 MJ

OVERALL LOSS COEFFICIENT = 0.493 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/09/85

TEST STARTING AT 20:50

TEST DURATION = 8 HOURS

INITIAL TEMP = 58.4 C

FINAL TEMP = 50.4 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	58.16	57.83	0.30	13.23	24.21
2	57.66	56.33	0.20	11.34	22.69
3	57.01	54.79	0.66	10.62	21.93
4	56.30	53.54	0.01	9.82	21.18
5	55.61	52.46	0.07	9.52	20.61
6	54.83	51.42	0.28	8.59	19.76
7	54.06	50.39	0.06	8.63	19.25
8	53.27	49.49	0.00	8.40	18.86

LOST ENERGY = 3.921 MJ

OVERALL LOSS COEFFICIENT = 2.651 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/10/85

TEST STARTING AT 21:15

TEST DURATION = 8 HOURS

INITIAL TEMP = 59.8 C

FINAL TEMP = 51.6 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	59.66	59.25	2.14	12.46	22.69
2	59.10	57.40	1.61	11.56	22.04
3	58.44	55.79	0.50	11.19	21.15
4	57.71	54.50	0.67	11.89	20.97
5	57.00	53.35	0.15	10.13	20.25
6	56.21	52.33	0.02	9.90	19.66
7	55.44	51.41	0.21	14.94	19.82
8	54.65	50.56	0.62	10.24	19.84

LOST ENERGY = 4.017 MJ

OVERALL LOSS COEFFICIENT = 2.590 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/13/85

TEST STARTING AT 22:10

TEST DURATION = 8 HOURS

INITIAL TEMP = 69.2 C

FINAL TEMP = 55.7 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	68.93	67.89	0.94	-11.98	9.44
2	68.03	64.55	0.12	-12.61	7.58
3	66.94	62.19	0.00	-13.64	5.93
4	65.76	60.22	0.33	-14.44	4.97
5	64.53	58.51	0.00	-14.94	4.30
6	63.28	56.87	0.00	-15.90	3.86
7	61.96	55.36	0.00	-16.21	3.15
8	60.52	54.03	0.00	-17.44	2.55

LOST ENERGY = 6.595 MJ

OVERALL LOSS COEFFICIENT = 2.593 W/(SQ M C)

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OUTDOOR STRATIFIED TEST, 09/14/85

TEST STARTING AT  1: 5

TEST DURATION =  8 HOURS

INITIAL TEMP  = 71.1 C

FINAL   TEMP  = 57.5 C


  HR     MEAS TANK TEMP     WIND        SKY         AMB
  #       TTOP     TBOT      VEL        TEMP        TEMP
          (C)      (C)      (M/S)       (C)         (C)

  1      70.90    69.74     0.15      -12.80        6.25
  2      69.87    65.69     0.01      -13.25        5.00
  3      68.72    63.63     0.00      -13.79        4.55
  4      67.46    61.69     0.00      -14.84        4.19
  5      66.21    59.84     0.00      -15.49        3.09
  6      64.86    58.10     0.01      -15.20        2.83
  7      63.54    56.54     0.04      -14.84        1.91
  8      62.63    55.37     0.26       -6.41        3.23

LOST ENERGY                 =      6.639 MJ

OVERALL LOSS COEFFICIENT    =      2.473 W/(SQ M C)
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OUTDOOR STRATIFIED TEST, 09/15/85

TEST STARTING AT 22:40

TEST DURATION = 8 HOURS

INITIAL TEMP = 46.3 C

FINAL TEMP = 38.4 C

HR #	MEAS TANK TEMP TTOP (C)	MEAS TANK TEMP TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	46.22	45.67	0.00	-7.94	9.38
2	45.74	44.18	0.00	-9.15	8.07
3	45.18	42.73	0.00	-10.15	7.18
4	44.52	41.45	0.00	-10.91	6.54
5	43.83	40.34	0.00	-10.15	6.03
6	43.09	39.30	0.00	-11.25	5.43
7	42.36	38.36	0.00	-10.63	4.71
8	41.56	37.48	0.00	-10.39	4.20

LOST ENERGY = 3.891 MJ

OVERALL LOSS COEFFICIENT = 2.434 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/16/85

TEST STARTING AT 22:40

TEST DURATION = 8 HOURS

INITIAL TEMP = 56.2 C

FINAL TEMP = 46.6 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	56.20	55.48	0.00	-5.69	10.27
2	55.60	53.55	0.00	-6.92	9.34
3	54.84	51.73	0.01	-7.25	8.71
4	54.01	50.15	0.00	-7.80	8.00
5	53.17	48.79	0.00	-8.87	7.31
6	52.24	47.58	0.00	-9.48	6.32
7	51.30	46.45	0.00	-10.01	6.10
8	50.36	45.36	0.00	-10.62	5.79

LOST ENERGY = 4.711 MJ

OVERALL LOSS COEFFICIENT = 2.425 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/17/85

TEST STARTING AT 22:45

TEST DURATION = 8 HOURS

INITIAL TEMP = 55.1 C

FINAL TEMP = 45.8 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	55.02	54.29	0.00	-3.86	11.53
2	54.42	52.48	0.00	-4.26	10.54
3	53.71	50.73	0.09	-4.53	10.56
4	52.92	49.31	0.00	-5.51	9.45
5	52.11	48.01	0.00	-6.38	9.06
6	51.28	46.80	0.00	-7.52	7.87
7	50.38	45.71	0.00	-7.76	7.63
8	49.44	44.65	0.00	-3.91	7.16

LOST ENERGY = 4.566 MJ

OVERALL LOSS COEFFICIENT = 2.490 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/18/85

TEST STARTING AT 21:50

TEST DURATION = 8 HOURS

INITIAL TEMP = 55.4 C

FINAL TEMP = 46.4 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	55.36	54.69	0.00	0.04	13.90
2	54.79	52.97	0.00	-0.72	12.87
3	54.06	51.31	0.00	-1.89	11.84
4	53.31	49.90	0.00	-2.76	11.11
5	52.53	48.66	0.00	-3.55	10.39
6	51.66	47.47	0.00	-3.37	9.75
7	50.80	46.37	0.00	-4.17	8.98
8	49.89	45.33	0.06	-5.42	8.27

LOST ENERGY = 4.417 MJ

OVERALL LOSS COEFFICIENT = 2.484 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/19/85

TEST STARTING AT 23:25

TEST DURATION = 8 HOURS

INITIAL TEMP = 55.0 C

FINAL TEMP = 45.8 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	54.88	54.20	0.00	0.81	13.79
2	54.28	52.45	0.00	0.93	13.03
3	53.55	50.78	0.00	0.43	12.84
4	52.74	49.33	0.00	-0.77	11.95
5	51.90	48.08	0.00	-0.94	11.31
6	50.98	46.92	0.00	-1.41	10.65
7	50.08	45.90	0.00	-1.93	10.31
8	49.14	44.86	0.00	-2.50	9.56

LOST ENERGY = 4.517 MJ

OVERALL LOSS COEFFICIENT = 2.627 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/20/85

TEST STARTING AT 22:30

TEST DURATION = 8 HOURS

INITIAL TEMP = 52.5 C

FINAL TEMP = 44.3 C

HR #	MEAS TANK TEMP TTOP (C)	MEAS TANK TEMP TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	52.48	51.84	0.00	-1.11	14.31
2	51.92	50.17	0.00	-2.58	13.12
3	51.24	48.62	0.00	-2.76	11.81
4	50.50	47.29	0.00	0.17	11.60
5	49.69	46.10	0.00	2.14	11.58
6	48.90	45.08	0.00	1.81	11.48
7	48.09	44.18	0.00	2.39	11.60
8	47.27	43.32	0.00	1.72	11.24

LOST ENERGY = 4.029 MJ

OVERALL LOSS COEFFICIENT = 2.497 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/21/85

TEST STARTING AT 0:50

TEST DURATION = 8 HOURS

INITIAL TEMP = 48.4 C

FINAL TEMP = 42.1 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	48.44	47.89	0.19	9.74	15.96
2	47.99	46.56	0.00	9.71	15.98
3	47.47	45.37	0.00	8.71	15.50
4	46.90	44.37	0.11	8.78	15.51
5	46.29	43.43	0.00	9.21	15.50
6	45.66	42.63	0.00	8.66	15.00
7	45.03	41.84	0.00	5.28	14.59
8	44.50	41.20	0.14	4.43	14.91

LOST ENERGY = 3.099 MJ

OVERALL LOSS COEFFICIENT = 2.335 W/(SQ M C)

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OUTDOOR STRATIFIED TEST, 09/22/85

TEST STARTING AT 22:25

TEST DURATION =  8 HOURS

INITIAL TEMP  = 45.7 C

FINAL   TEMP  = 39.5 C


  HR     MEAS TANK TEMP      WIND       SKY        AMB
  #        TTOP     TBOT      VEL       TEMP       TEMP
           (C)      (C)      (M/S)      (C)        (C)

  1      45.72     45.23      0.00       2.80      14.82
  2      45.30     44.00      0.00       4.14      14.20
  3      44.80     42.79      0.00       2.27      14.07
  4      44.23     41.75      0.00       1.77      13.27
  5      43.62     40.87      0.00       5.69      13.15
  6      43.03     40.07      0.00      11.26      14.41
  7      42.46     39.41      0.06      10.73      14.32
  8      41.86     38.74      0.17      11.68      14.27

LOST ENERGY                  =      3.053 MJ

OVERALL LOSS COEFFICIENT     =      2.409 W/(SQ M C)
```

OUTDOOR STRATIFIED TEST, 09/24/85

TEST STARTING AT 21:55

TEST DURATION = 8 HOURS

INITIAL TEMP = 45.0 C

FINAL TEMP = 38.9 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	45.16	44.43	1.71	-4.26	13.30
2	44.64	43.40	0.36	-5.52	11.91
3	44.09	42.44	0.17	-5.51	10.34
4	43.49	41.56	0.07	-4.93	9.53
5	42.84	40.74	0.01	-3.37	8.95
6	42.18	39.97	0.02	-6.29	8.23
7	41.50	39.22	0.00	-6.65	7.54
8	40.83	38.47	0.03	-7.62	7.17

LOST ENERGY = 3.005 MJ

OVERALL LOSS COEFFICIENT = 2.087 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 09/25/85

TEST STARTING AT 21:30

TEST DURATION = 8 HOURS

INITIAL TEMP = 47.8 C

FINAL TEMP = 41.5 C

HR #	MEAS TANK TEMP TTOP (C)	MEAS TANK TEMP TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	47.87	47.30	0.65	12.12	15.60
2	47.45	46.02	0.25	11.45	15.39
3	46.97	44.81	0.87	11.71	15.04
4	46.40	43.77	0.63	11.60	14.77
5	45.85	42.86	0.25	11.22	14.53
6	45.25	42.06	0.45	10.89	14.37
7	44.64	41.31	0.06	10.62	14.24
8	44.06	40.68	0.05	10.39	14.18

LOST ENERGY = 3.100 MJ

OVERALL LOSS COEFFICIENT = 2.335 W/(SQ M C)

OUTDOOR FORCED TEST, 09/26/85

TEST STARTING AT 22:20

TEST DURATION = 8 HOURS

INITIAL TEMP = 52.6 C

FINAL TEMP = 49.9 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	52.62	52.38	1.20	7.50	17.35
2	52.33	52.10	2.08	10.67	16.93
3	52.00	51.79	2.60	11.28	16.78
4	51.69	51.46	2.59	8.08	15.32
5	51.36	51.12	3.01	9.97	14.87
6	51.00	50.76	2.76	8.11	14.02
7	50.61	50.37	3.38	3.37	12.84
8	50.22	49.97	4.12	1.39	12.23

LOST ENERGY = 1.325 MJ

OVERALL LOSS COEFFICIENT = 0.819 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 10/01/85

TEST STARTING AT 23: 0

TEST DURATION = 8 HOURS

INITIAL TEMP = 59.5 C

FINAL TEMP = 49.4 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.33	58.63	0.01	10.58	15.72
2	58.63	56.60	0.01	10.47	15.31
3	57.78	54.82	0.00	10.81	15.36
4	56.84	53.35	0.01	11.11	16.35
5	55.87	52.05	0.00	11.45	16.59
6	54.90	50.89	0.07	11.97	16.59
7	53.90	49.73	0.85	11.79	16.35
8	52.89	48.53	0.98	10.89	15.26

LOST ENERGY = 4.950 MJ

OVERALL LOSS COEFFICIENT = 2.893 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 10/08/85

TEST STARTING AT 23: 0

TEST DURATION = 8 HOURS

INITIAL TEMP = 48.3 C

FINAL TEMP = 40.7 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	48.16	47.86	0.13	7.08	14.23
2	47.72	46.47	0.06	7.49	13.59
3	47.17	45.15	0.00	0.84	12.57
4	46.54	43.98	0.00	-2.50	10.87
5	45.78	42.83	0.00	-3.86	9.92
6	45.05	41.80	0.00	-5.15	8.97
7	44.31	40.84	0.00	-5.56	8.15
8	43.50	39.90	0.00	-6.47	7.81

LOST ENERGY = 3.740 MJ

OVERALL LOSS COEFFICIENT = 2.497 W/(SQ M C)

OUTDOOR STRATIFIED TEST, 10/10/85

TEST STARTING AT 23: 0

TEST DURATION = 8 HOURS

INITIAL TEMP = 61.5 C

FINAL TEMP = 51.1 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	61.32	60.67	0.00	2.63	13.73
2	60.63	58.59	0.00	4.76	13.94
3	59.77	56.71	0.00	1.80	13.77
4	58.82	55.12	0.00	0.76	13.21
5	57.84	53.69	0.00	-0.04	12.48
6	56.80	52.41	0.00	-0.85	11.96
7	55.73	51.18	0.00	1.26	11.62
8	54.70	50.03	0.00	4.63	12.21

LOST ENERGY = 5.094 MJ

OVERALL LOSS COEFFICIENT = 2.642 W/(SQ M C)

SRCC TYPE TEST, 10/22/85

TEST STARTING AT 15:30

TEST DURATION = 16 HOURS

INITIAL TEMP = 60.5 C

FINAL TEMP = 47.2 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	60.35	60.16	5.65	21.38	22.63
2	59.76	59.09	5.59	21.56	22.79
3	59.05	57.86	5.63	21.32	22.54
4	58.29	56.67	5.54	21.57	22.71
5	57.49	55.58	5.53	21.65	22.79
6	56.68	54.56	5.70	21.49	22.65
7	55.85	53.60	5.67	21.47	22.63
8	55.03	52.71	5.44	21.44	22.61
9	54.21	51.86	5.60	21.47	22.66
10	53.40	51.04	5.68	21.44	22.65
11	52.61	50.27	5.54	21.41	22.60
12	51.84	49.51	5.59	21.42	22.63
13	51.08	48.80	5.59	21.42	22.61
14	50.34	48.09	5.65	21.46	22.66
15	49.63	47.43	5.52	21.63	22.79
16	48.93	46.78	5.56	21.53	22.68

LOST ENERGY = 6.520 MJ

OVERALL LOSS COEFFICIENT = 2.381 W/(SQ M C)

```
SRCC TYPE TEST, 10/23/85

TEST STARTING AT 12: 5

TEST DURATION = 16 HOURS

INITIAL TEMP  = 59.8 C

FINAL   TEMP  = 46.0 C


  HR     MEAS TANK TEMP      WIND       SKY       AMB
  #      TTOP     TBOT       VEL       TEMP      TEMP
          (C)      (C)      (M/S)       (C)       (C)

  1      59.61    59.43      5.64     10.17     22.05
  2      58.98    58.34      5.69     10.96     22.16
  3      58.23    57.07      5.51     11.59     22.41
  4      57.42    55.85      5.73     11.83     22.29
  5      56.57    54.73      5.63     12.09     22.23
  6      55.70    53.68      5.58     12.37     22.32
  7      54.82    52.71      5.68     12.41     22.28
  8      53.96    51.78      5.64     12.44     22.22
  9      53.10    50.91      5.74     12.59     22.34
 10      52.26    50.07      5.73     12.54     22.30
 11      51.43    49.27      5.62     12.51     22.23
 12      50.63    48.50      5.55     12.69     22.39
 13      49.85    47.75      5.62     12.67     22.36
 14      49.09    47.04      5.67     12.56     22.20
 15      48.35    46.35      5.66     12.75     22.36
 16      47.64    45.67      5.90     12.74     22.36

LOST ENERGY                =       6.768 MJ

OVERALL LOSS COEFFICIENT   =       2.521 W/(SQ M C)
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SRCC TYPE TEST, 10/24/85

TEST STARTING AT 11: 0

TEST DURATION = 16 HOURS

INITIAL TEMP  = 61.3 C

FINAL   TEMP  = 46.6 C


 HR     MEAS TANK TEMP      WIND      SKY       AMB
 #      TTOP      TBOT      VEL       TEMP      TEMP
        (C)       (C)       (M/S)     (C)       (C)

 1      61.12     60.95     0.00      1.08      22.31
 2      60.42     59.82     0.00      0.82      22.19
 3      59.60     58.54     0.00      0.83      22.21
 4      58.72     57.29     0.00      0.86      22.26
 5      57.80     56.13     0.00      0.85      22.23
 6      56.86     55.04     0.00      0.78      22.01
 7      55.92     54.01     0.00      0.84      22.11
 8      54.99     53.03     0.00      0.85      22.10
 9      54.07     52.09     0.00      0.86      22.10
10      53.16     51.18     0.00      0.85      22.08
11      52.27     50.31     0.00      0.85      22.06
12      51.40     49.48     0.00      0.81      22.01
13      50.56     48.66     0.00      0.79      22.00
14      49.73     47.86     0.00      0.74      21.94
15      48.93     47.09     0.00      0.66      21.87
16      48.14     46.36     0.00      0.64      21.87

LOST ENERGY                  =      7.206 MJ

OVERALL LOSS COEFFICIENT     =      2.581 W/(SQ M C)
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SRCC TYPE TEST, 10/25/85

TEST STARTING AT 17:35

TEST DURATION = 16 HOURS

INITIAL TEMP = 60.8 C

FINAL TEMP = 48.4 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	60.65	60.48	0.00	23.17	21.82
2	60.09	59.52	0.00	23.08	21.56
3	59.42	58.45	0.00	23.10	21.89
4	58.69	57.40	0.00	22.98	21.69
5	57.95	56.42	0.00	23.15	21.99
6	57.18	55.51	0.00	23.08	21.89
7	56.41	54.65	0.00	22.93	21.76
8	55.65	53.84	0.00	22.79	21.62
9	54.88	53.05	0.00	22.63	21.47
10	54.13	52.29	0.00	22.46	21.35
11	53.38	51.55	0.00	22.30	21.21
12	52.66	50.83	0.00	22.14	21.08
13	51.94	50.14	0.00	21.97	20.94
14	51.23	49.46	0.00	21.80	20.80
15	50.55	48.80	0.00	21.64	20.68
16	49.87	48.15	0.00	21.52	20.68

LOST ENERGY = 6.077 MJ

OVERALL LOSS COEFFICIENT = 2.075 W/(SQ M C)

SRCC TYPE TEST, 10/28/85

TEST STARTING AT 18:30

TEST DURATION = 16 HOURS

INITIAL TEMP = 62.3 C

FINAL TEMP = 50.3 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	62.08	61.94	0.00	32.23	21.79
2	61.56	61.01	0.00	32.60	21.77
3	60.92	59.96	0.00	32.87	21.71
4	60.23	58.94	0.00	32.86	21.62
5	59.51	57.99	0.00	32.31	21.52
6	58.78	57.10	0.00	31.40	21.43
7	58.03	56.26	0.00	30.28	21.33
8	57.28	55.46	0.00	29.74	21.21
9	56.55	54.70	0.00	29.35	21.15
10	55.82	53.97	0.00	29.69	21.02
11	55.10	53.26	0.00	29.66	20.90
12	54.41	52.57	0.00	30.12	20.77
13	53.72	51.91	0.00	30.15	20.68
14	53.05	51.26	0.00	30.16	20.63
15	52.39	50.63	0.00	30.55	20.66
16	51.76	50.01	0.00	30.61	20.90

LOST ENERGY = 5.877 MJ

OVERALL LOSS COEFFICIENT = 1.894 W/(SQ M C)

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SRCC TYPE TEST, 10/31/85

TEST STARTING AT 13: 0

TEST DURATION = 16 HOURS

INITIAL TEMP  = 63.1 C

FINAL   TEMP  = 49.4 C


  HR     MEAS TANK TEMP      WIND       SKY        AMB
  #       TTOP     TBOT       VEL       TEMP       TEMP
          (C)      (C)       (M/S)      (C)        (C)

  1      62.86    62.68      4.27      28.60      24.30
  2      62.22    61.61      4.26      28.68      24.44
  3      61.46    60.37      4.36      28.73      24.55
  4      60.64    59.18      4.35      28.73      24.57
  5      59.78    58.07      4.20      28.84      24.61
  6      58.90    57.04      4.33      28.86      24.64
  7      58.03    56.08      3.97      28.83      24.70
  8      57.16    55.17      4.33      28.78      24.76
  9      56.30    54.29      4.23      28.78      24.81
 10      55.47    53.46      4.11      28.83      24.85
 11      54.64    52.68      4.42      28.85      24.87
 12      53.85    51.92      4.31      28.85      24.89
 13      53.08    51.18      4.17      28.85      24.90
 14      52.33    50.47      4.20      28.82      24.92
 15      51.60    49.79      4.37      28.78      24.95
 16      50.89    49.13      4.32      28.75      24.97

LOST ENERGY                  =      6.710 MJ

OVERALL LOSS COEFFICIENT     =      2.421 W/(SQ M C)
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SRCC TYPE TEST, 11/11/85

TEST STARTING AT 20:55

TEST DURATION = 16 HOURS

INITIAL TEMP = 59.9 C

FINAL TEMP = 46.0 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.74	59.56	5.35	13.11	23.23
2	59.12	58.48	5.37	13.05	23.11
3	58.35	57.22	5.44	12.88	23.02
4	57.53	56.01	5.47	12.85	22.92
5	56.65	54.89	5.43	12.85	22.85
6	55.78	53.84	5.70	12.78	22.77
7	54.88	52.85	5.48	12.72	22.71
8	54.00	51.92	5.58	12.70	22.67
9	53.13	51.03	5.48	12.67	22.58
10	52.27	50.19	5.44	12.60	22.49
11	51.43	49.37	5.48	12.56	22.42
12	50.61	48.57	5.56	12.55	22.38
13	49.81	47.81	5.41	12.55	22.40
14	49.03	47.08	5.46	12.59	22.44
15	48.29	46.37	5.43	12.63	22.55
16	47.56	45.69	5.46	12.72	22.72

LOST ENERGY = 6.817 MJ

OVERALL LOSS COEFFICIENT = 2.570 W/(SQ M C)

SRCC TYPE TEST, 11/14/85

TEST STARTING AT 12: 0

TEST DURATION = 16 HOURS

INITIAL TEMP = 59.9 C

FINAL TEMP = 46.8 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	59.73	59.56	0.00	12.82	22.79
2	59.12	58.57	0.00	12.84	22.89
3	58.38	57.44	0.00	12.84	23.01
4	57.60	56.34	0.00	12.76	22.74
5	56.78	55.32	0.00	12.82	22.82
6	55.95	54.35	0.00	12.77	22.75
7	55.12	53.44	0.00	12.72	22.65
8	54.29	52.57	0.00	12.71	22.61
9	53.47	51.73	0.00	12.73	22.69
10	52.66	50.93	0.00	12.68	22.49
11	51.86	50.15	0.00	12.63	22.40
12	51.09	49.40	0.00	12.66	22.51
13	50.34	48.67	0.00	12.64	22.41
14	49.60	47.96	0.00	12.68	22.52
15	48.88	47.27	0.00	12.72	22.45
16	48.19	46.62	0.00	12.75	22.60

LOST ENERGY = 6.424 MJ

OVERALL LOSS COEFFICIENT = 2.384 W/(SQ M C)

INDOOR FORCED TEST, 09/12/85

TEST STARTING AT 12:55

TEST DURATION = 8 HOURS

INITIAL TEMP = 53.1 C

FINAL TEMP = 50.4 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	52.99	52.98	0.00	7.51	21.82
2	52.63	52.62	0.00	7.22	21.25
3	52.26	52.25	0.00	7.16	21.03
4	51.90	51.89	0.00	7.12	20.92
5	51.54	51.53	0.00	7.12	20.80
6	51.19	51.19	0.00	7.13	20.69
7	50.86	50.85	0.00	7.11	20.58
8	50.53	50.51	0.00	7.08	20.50

LOST ENERGY = 1.325 MJ

OVERALL LOSS COEFFICIENT = 0.966 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/13/85

TEST STARTING AT 13:25

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 53.6 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.88	59.71	0.00	22.33	20.52
2	59.37	58.79	0.00	22.20	20.43
3	58.77	57.72	0.00	22.10	20.38
4	58.13	56.68	0.00	22.05	20.44
5	57.46	55.71	0.00	21.93	20.40
6	56.78	54.81	0.00	21.87	20.39
7	56.07	53.97	0.00	21.78	20.35
8	55.36	53.17	0.00	21.69	20.26

LOST ENERGY = 3.134 MJ

OVERALL LOSS COEFFICIENT = 1.934 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/14/85

TEST STARTING AT 15:45

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 52.9 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.80	59.64	0.00	9.72	20.79
2	59.22	58.64	0.00	9.68	20.95
3	58.55	57.48	0.00	9.59	21.03
4	57.84	56.35	0.00	9.59	20.98
5	57.09	55.30	0.00	9.53	20.74
6	56.31	54.32	0.00	9.59	20.98
7	55.53	53.41	0.00	9.48	20.61
8	54.74	52.55	0.00	9.56	20.84

LOST ENERGY = 3.477 MJ

OVERALL LOSS COEFFICIENT = 2.198 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/15/85

TEST STARTING AT 13:15

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 52.4 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	59.76	59.59	0.00	1.65	20.43
2	59.13	58.53	0.00	1.58	20.36
3	58.41	57.29	0.00	1.58	20.40
4	57.63	56.09	0.00	1.53	20.43
5	56.82	54.98	0.00	1.51	20.41
6	55.99	53.94	0.00	1.53	20.38
7	55.15	52.96	0.00	1.56	20.28
8	54.30	52.02	0.00	1.59	20.21

LOST ENERGY = 3.722 MJ

OVERALL LOSS COEFFICIENT = 2.338 W/(SQ M C)

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INDOOR STRATIFIED TEST, 09/16/85

TEST STARTING AT 14: 5

TEST DURATION =  8 HOURS

INITIAL TEMP  = 60.0 C

FINAL   TEMP  = 52.0 C


  HR    MEAS TANK TEMP    WIND     SKY      AMB
  #      TTOP     TBOT     VEL     TEMP     TEMP
          (C)      (C)    (M/S)     (C)      (C)

  1      59.77    59.59    0.00    -6.84    19.71
  2      59.09    58.45    0.00    -6.70    19.88
  3      58.33    57.15    0.00    -6.56    19.97
  4      57.50    55.89    0.00    -6.43    19.86
  5      56.65    54.71    0.00    -6.19    19.91
  6      55.76    53.62    0.00    -6.02    19.82
  7      54.88    52.59    0.00    -6.05    19.71
  8      53.99    51.62    0.00    -6.01    19.81

LOST ENERGY                 =       3.919 MJ

OVERALL LOSS COEFFICIENT    =       2.435 W/(SQ M C)
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INDOOR STRATIFIED TEST, 09/17/85

TEST STARTING AT 14:10

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 52.3 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.78	59.60	2.28	3.75	19.80
2	59.16	58.48	2.16	4.16	19.83
3	58.45	57.20	2.23	4.58	20.01
4	57.69	55.96	2.30	4.73	19.86
5	56.89	54.81	2.49	5.10	19.86
6	56.06	53.76	2.37	5.39	19.87
7	55.24	52.76	2.31	5.54	19.82
8	54.40	51.84	2.26	5.50	19.79

LOST ENERGY = 3.771 MJ

OVERALL LOSS COEFFICIENT = 2.340 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/18/85

TEST STARTING AT 11:30

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 52.4 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.83	59.65	2.13	5.81	19.94
2	59.21	58.55	2.17	5.66	19.99
3	58.51	57.26	2.23	5.74	20.04
4	57.75	56.02	2.25	5.84	20.09
5	56.95	54.88	2.18	5.93	20.11
6	56.14	53.83	2.14	6.09	20.17
7	55.30	52.85	2.26	5.93	19.77
8	54.47	51.92	2.37	5.85	19.50

LOST ENERGY = 3.722 MJ

OVERALL LOSS COEFFICIENT = 2.312 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/19/85

TEST STARTING AT 12:10

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 52.4 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.82	59.65	2.20	8.86	20.40
2	59.21	58.56	2.23	8.69	20.18
3	58.51	57.29	2.23	8.57	19.87
4	57.74	56.07	2.13	8.54	19.77
5	56.95	54.93	2.26	8.56	19.72
6	56.13	53.87	2.20	8.59	19.70
7	55.30	52.89	2.27	8.67	19.65
8	54.46	51.95	2.20	8.65	19.56

LOST ENERGY = 3.722 MJ

OVERALL LOSS COEFFICIENT = 2.306 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/20/85

TEST STARTING AT 13:55

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 52.7 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	59.82	59.67	2.14	19.50	20.69
2	59.25	58.64	2.10	19.13	20.30
3	58.57	57.43	2.16	19.02	20.19
4	57.84	56.24	2.16	18.96	20.12
5	57.07	55.15	2.23	18.88	20.05
6	56.29	54.13	2.25	18.82	19.97
7	55.48	53.17	2.21	18.71	19.85
8	54.67	52.26	2.23	18.57	19.71

LOST ENERGY = 3.575 MJ

OVERALL LOSS COEFFICIENT = 2.217 W/(SQ M C)

INDOOR FORCED TEST, 09/21/85

TEST STARTING AT 16:15

TEST DURATION = 8 HOURS

INITIAL TEMP = 60.0 C

FINAL TEMP = 54.9 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	59.68	59.66	0.00	-4.15	19.53
2	58.98	58.96	0.00	-4.12	19.35
3	58.30	58.28	0.00	-3.91	19.32
4	57.64	57.62	0.00	-3.54	19.36
5	57.00	56.99	0.00	-3.33	19.37
6	56.38	56.37	0.00	-3.17	19.38
7	55.78	55.77	0.00	-2.97	19.37
8	55.20	55.19	0.00	-2.85	19.32

LOST ENERGY = 2.497 MJ

OVERALL LOSS COEFFICIENT = 1.472 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/23/85

TEST STARTING AT 13:30

TEST DURATION = 8 HOURS

INITIAL TEMP = 83.9 C

FINAL TEMP = 70.9 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	83.53	83.31	0.00	0.20	20.67
2	82.34	81.49	0.00	0.23	20.75
3	81.00	79.40	0.00	0.28	20.79
4	79.59	77.38	0.00	0.22	20.48
5	78.13	75.51	0.00	0.22	20.38
6	76.65	73.76	0.00	0.26	20.29
7	75.17	72.11	0.00	0.27	20.21
8	73.70	70.55	0.00	0.26	20.15

LOST ENERGY = 6.313 MJ

OVERALL LOSS COEFFICIENT = 2.496 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/24/85

TEST STARTING AT 13:15

TEST DURATION = 8 HOURS

INITIAL TEMP = 87.2 C

FINAL TEMP = 73.4 C

HR #	MEAS TANK TEMP TTOP (C)	MEAS TANK TEMP TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	86.73	86.49	0.00	2.74	21.49
2	85.44	84.57	0.00	2.68	21.45
3	84.00	82.38	0.00	2.60	21.37
4	82.48	80.25	0.00	2.55	21.49
5	80.91	78.29	0.00	2.41	21.32
6	79.33	76.46	0.00	2.30	21.18
7	77.75	74.72	0.00	2.21	21.05
8	76.18	73.08	0.00	2.14	20.96

LOST ENERGY = 6.693 MJ

OVERALL LOSS COEFFICIENT = 2.553 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/25/85

TEST STARTING AT 12:45

TEST DURATION = 8 HOURS

INITIAL TEMP = 59.9 C

FINAL TEMP = 52.3 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	59.77	59.51	0.00	3.51	19.59
2	59.14	58.42	0.00	3.57	19.68
3	58.40	57.20	0.00	3.62	19.79
4	57.61	55.99	0.00	3.69	19.90
5	56.79	54.88	0.00	3.73	19.94
6	55.95	53.85	0.00	3.75	19.95
7	55.10	52.88	0.00	3.72	19.90
8	54.24	51.96	0.00	3.69	19.88

LOST ENERGY = 3.723 MJ

OVERALL LOSS COEFFICIENT = 2.306 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/26/85

TEST STARTING AT 11:50

TEST DURATION = 8 HOURS

INITIAL TEMP = 68.5 C

FINAL TEMP = 59.2 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	68.24	68.01	0.00	4.01	20.09
2	67.43	66.68	0.00	4.11	20.31
3	66.53	65.18	0.00	4.15	20.43
4	65.56	63.72	0.00	4.20	20.50
5	64.55	62.38	0.00	4.25	20.55
6	63.52	61.12	0.00	4.16	20.22
7	62.48	59.94	0.00	4.10	20.02
8	61.45	58.81	0.00	4.01	19.84

LOST ENERGY = 4.541 MJ

OVERALL LOSS COEFFICIENT = 2.356 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/27/85

TEST STARTING AT 13:10

TEST DURATION = 8 HOURS

INITIAL TEMP = 65.9 C

FINAL TEMP = 56.9 C

HR #	MEAS TANK TEMP		WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
	TTOP (C)	TBOT (C)			
1	65.61	65.42	0.00	5.25	19.22
2	64.87	64.17	0.00	5.17	19.14
3	64.01	62.73	0.00	5.18	19.22
4	63.09	61.35	0.00	5.13	18.93
5	62.13	60.05	0.00	4.94	18.45
6	61.14	58.83	0.00	4.87	18.18
7	60.15	57.68	0.00	4.76	17.94
8	59.14	56.58	0.00	4.73	17.75

LOST ENERGY = 4.399 MJ

OVERALL LOSS COEFFICIENT = 2.311 W/(SQ M C)

INDOOR STRATIFIED TEST, 09/28/85

TEST STARTING AT 13:25

TEST DURATION = 8 HOURS

INITIAL TEMP = 64.5 C

FINAL TEMP = 56.0 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	64.28	64.10	0.00	6.15	19.33
2	63.56	62.88	0.00	6.07	19.19
3	62.73	61.49	0.00	6.02	19.16
4	61.84	60.15	0.00	5.98	19.13
5	60.91	58.90	0.00	6.01	19.15
6	59.96	57.74	0.00	5.99	19.13
7	59.01	56.64	0.00	5.98	19.03
8	58.05	55.61	0.00	5.95	18.97

LOST ENERGY = 4.156 MJ

OVERALL LOSS COEFFICIENT = 2.271 W/(SQ M C)

INDOOR FORCED TEST, 11/03/85

TEST STARTING AT 20:45

TEST DURATION = 8 HOURS

INITIAL TEMP = 54.3 C

FINAL TEMP = 51.0 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	58.11	57.94	0.23	10.04	21.05
2	57.49	56.91	0.26	10.00	20.94
3	56.74	55.76	0.18	9.97	20.90
4	55.95	54.63	0.20	9.92	20.81
5	55.11	53.58	0.22	9.90	20.73
6	54.26	52.59	0.15	9.87	20.63
7	53.41	51.65	0.19	9.83	20.51
8	52.55	50.76	0.15	9.84	20.45

LOST ENERGY = 1.619 MJ

OVERALL LOSS COEFFICIENT = 1.158 W/(SQ M C)

INDOOR STRATIFIED TEST, 11/04/85

TEST STARTING AT 15:35

TEST DURATION = 8 HOURS

INITIAL TEMP = 53.5 C

FINAL TEMP = 47.7 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	53.37	53.22	0.00	16.34	21.67
2	52.89	52.42	0.00	16.34	21.71
3	52.31	51.51	0.00	16.31	21.77
4	51.68	50.61	0.00	16.24	21.66
5	51.02	49.78	0.00	16.17	21.53
6	50.36	48.99	0.00	16.10	21.24
7	49.68	48.23	0.00	16.04	21.10
8	49.00	47.52	0.00	15.99	21.08

LOST ENERGY = 2.847 MJ

OVERALL LOSS COEFFICIENT = 2.199 W/(SQ M C)

INDOOR STRATIFIED TEST, 11/05/85

TEST STARTING AT 11:35

TEST DURATION = 8 HOURS

INITIAL TEMP = 53.1 C

FINAL TEMP = 47.2 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	52.99	52.85	0.00	12.70	21.22
2	52.50	52.02	0.00	12.65	21.24
3	51.91	51.09	0.00	12.64	21.34
4	51.27	50.18	0.00	12.53	21.14
5	50.60	49.32	0.00	12.51	21.12
6	49.92	48.51	0.00	12.53	21.26
7	49.22	47.75	0.00	12.44	21.03
8	48.54	47.02	0.00	12.42	21.04

LOST ENERGY = 2.897 MJ

OVERALL LOSS COEFFICIENT = 2.249 W/(SQ M C)

INDOOR STRATIFIED TEST, 11/07/85

TEST STARTING AT 12:30

TEST DURATION = 8 HOURS

INITIAL TEMP = 55.6 C

FINAL TEMP = 49.1 C

HR	MEAS TANK TEMP		WIND	SKY	AMB
#	TTOP (C)	TBOT (C)	VEL (M/S)	TEMP (C)	TEMP (C)
1	55.49	55.34	0.37	5.00	22.41
2	54.94	54.43	0.36	4.87	22.47
3	54.28	53.39	0.30	4.94	22.46
4	53.57	52.37	0.35	4.97	22.37
5	52.83	51.42	0.39	4.95	22.30
6	52.06	50.52	0.38	4.87	22.15
7	51.30	49.69	0.43	5.02	22.25
8	50.54	48.88	0.32	4.98	22.21

LOST ENERGY = 3.189 MJ

OVERALL LOSS COEFFICIENT = 2.386 W/(SQ M C)

INDOOR STRATIFIED TEST, 11/12/85

TEST STARTING AT 15:30

TEST DURATION = 8 HOURS

INITIAL TEMP = 59.9 C

FINAL TEMP = 52.4 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	59.68	59.51	0.90	11.49	21.55
2	59.04	58.45	0.92	11.38	21.46
3	58.27	57.27	0.93	11.37	21.38
4	57.44	56.11	0.93	11.33	21.32
5	56.58	55.02	0.90	11.26	21.22
6	55.71	54.01	0.93	11.24	21.21
7	54.83	53.05	1.04	11.22	21.17
8	53.97	52.14	0.81	11.08	21.15

LOST ENERGY = 3.673 MJ

OVERALL LOSS COEFFICIENT = 2.370 W/(SQ M C)

INDOOR STRATIFIED TEST, 11/13/85

TEST STARTING AT 10:15

TEST DURATION = 8 HOURS

INITIAL TEMP = 58.3 C

FINAL TEMP = 51.0 C

HR #	MEAS TANK TEMP TTOP (C)	TBOT (C)	WIND VEL (M/S)	SKY TEMP (C)	AMB TEMP (C)
1	54.07	54.06	0.27	10.04	20.95
2	53.60	53.59	0.28	10.29	21.22
3	53.15	53.14	0.39	10.41	21.45
4	52.73	52.71	0.29	10.35	21.50
5	52.31	52.30	0.26	10.30	21.41
6	51.90	51.90	0.32	10.31	21.32
7	51.52	51.51	0.26	10.24	21.23
8	51.14	51.13	0.37	9.85	21.15

LOST ENERGY = 3.578 MJ

OVERALL LOSS COEFFICIENT = 2.377 W/(SQ M C)