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**IN-SITU PARAMETER ESTIMATION FOR SOLAR DOMESTIC
HOT WATER HEATING SYSTEMS COMPONENTS**

FINAL REPORT FOR THE PERIOD

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Toni R. Smith

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ABSTRACT OF DISSERTATION

IN-SITU PARAMETER ESTIMATION FOR SOLAR DOMESTIC HOT WATER HEATING SYSTEMS COMPONENTS

Three different solar domestic hot water systems are being tested at the Colorado State University Solar Energy Applications Laboratory; an unpressurized drain-back system with a load side heat exchanger, an integral collector storage system, and an ultra low flow natural convection heat exchanger system. The systems are fully instrumented to yield data appropriate for in-depth analyses of performance. The level of detail allows the observation of the performance of the total system and the performance of the individual components.

This dissertation evaluates the systems based on in-situ experimental data and compares the performances with simulated performances. The verification of the simulations aids in the rating procedure. The whole system performance measurements are also used to analyze the performance of individual components of a solar hot water system and to develop improved component models. The data are analyzed extensively and the parameters needed to characterize the systems fully are developed. Also resulting from this in-depth analysis are suggested design improvements either to the systems or the system components.

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Table of Contents

Abstract	iii
Acknowledgements	vi
Table of Contents	v
List of Tables	ix
List of Figures	x
Nomenclature	xii
 CHAPTER 1. INTRODUCTION	 1
1.1 Introduction	1
1.2 Background	3
1.3 Objective	5
1.4 Organization of Dissertation	6
 CHAPTER 2. DESCRIPTION OF SYSTEMS AND TESTS.....	 8
2.1 Introduction	8
2.2 Description of Systems	8
2.2.1 Overview of Systems	9
2.2.2 Ambient Weather Station	10
2.2.3 Drain-Back System	11
2.2.4 Integral Collector Storage System	13
2.2.5 Natural Convection Heat Exchanger System	15
2.3 Test Procedures	17
2.3.1 Normal Operation	18
2.3.2 Continuous Operation	19
2.3.3 Heat Loss Tests	19

2.4 Summary	19
 CHAPTER 3. EXPERIMENTAL APPROACH AND TESTING METHODOLOGY	 20
3.1 Introduction	20
3.2 Energy Balances	21
3.2.1 Convective Energy.....	21
3.2.2 Parasitic Power	22
3.2.3 Internal Energy	22
3.2.4 Overall System Energy	23
3.3 Instrumentation	24
3.3.1 Calibrations	24
3.3.2 Ambient Weather Station	27
3.3.3 Temperature Measurements	28
3.3.4 Fluid Flow Rate	30
3.3.5 Differential Pressure Measurements	32
3.3.6 Electrical Power Measurements	32
3.4 Overall Energy Balance Results	33
3.5 Test Revisions	36
3.6 Summary	38
 CHAPTER 4. STORAGE TANK HEAT LOSS TESTS.....	 39
4.1 Introduction	39
4.2 Heat Loss Test Method	39
4.2.1 Experimental Formulation.....	39
4.2.1.1 Area Weighting vs. Volume Weighting	42
4.2.2 Analytical Formulation	43
4.2.3 Valves Open/Valves Closed	44
4.2.4 Heat Traps	45
4.3 Results	49
4.4 Conclusions	54

CHAPTER 5. HEAT EXCHANGER ANALYSES	55
5.1 Introduction	55
5.2 Drain-Back System Load Side Heat Exchanger	55
5.2.1 Heat Exchanger Description	56
5.2.2 Experimental Results	58
5.2.3 Analysis	59
5.2.3.1 Effectiveness	60
5.2.3.2 Heat Transfer Coefficient Models	62
5.2.3.3 Experimental Data Reduction	66
5.2.3.4 Curve Fits	70
5.3 TRNSYS Simulation Results	72
5.4 Conclusions	74
CHAPTER 6. SOLAR COLLECTOR PARAMETER ESTIMATION	76
6.1 Introduction	76
6.1.1 General Collector Equation	77
6.1.2 Collector Capacitance	78
6.2 Problem Identification	80
6.3 Theoretical Formulation	82
6.3.1 Radiation Processor	83
6.3.2 Tau Alpha Product	86
6.3.3 $F_R U_L$	88
6.4 Approach	89
6.4.1 Absorbed Radiation	90
6.4.2 Tau Alpha Correlations	91
6.4.3 Heat Loss Models	94
6.4.4 Parametric Equations	95
6.4.4.1 Performance Equations in S	96
6.4.4.2 Model for $F_R U_L$	99
6.4.4.3 Combined Models	99
6.4.5 Least Squares	99
6.4.6 Quality of Fit Metrics	100
6.4.6.1 Radiation Processor	101
6.4.6.2 Collector Performance Equations	102
6.4.7 Possible Runs	102

6.4.8 Night Time Data Analysis	103
6.5 Results	105
6.5.1 Radiation Processor	105
6.5.1.1 Reindl Horizontal Model	108
6.5.1.2 Discussion	111
6.5.2 Parameter Estimation	112
6.5.2.1 Determination of All Parameters	112
6.5.2.2 Efficiency and Useful Energy Collector Equation	115
6.5.2.3 Determination of $F_R U_L$ from Night-Time Data	117
6.5.2.4 Determination of Optical Parameters with Known $F_R U_L$	121
6.5.2.5 $F_R U_L$ as a Linear Function of Wind Speed	121
6.5.3 Comparison of Predictions	124
6.5.3.1 Comparison to SRCC	125
6.5.3.2 Comparison to Second Data Set	128
6.6 Summary	131
CHAPTER 7 CONCLUSIONS AND RECOMMENDATIONS	133
7.1 Introduction	133
7.2 Conclusions	133
7.2.1 General	133
7.2.2 System Specific	135
7.3 Recommendations	137
References	139
Appendix A Instrumentation List	144
Appendix B Error Analysis	147
Appendix C Program Listings	154
Appendix D $F_R U_L$ Calculation	187

List of Tables

2.1 System Details	9
3.1 Instrumentation	27
3.2 Energy Balance Results	34
4.1 UA Test Conditions	49
4.2 UA Results	50
4.3 UA Results - Valves Closed Only	50
4.4 UA Results - Valves Open Only	51
4.5 UA Results - Valves Open/Closed Comparison	52
5.1 Heat Transfer Properties	69
5.2 Heat Exchanger Coefficients	71
5.3 Measured and TRNSYS Results	73
6.1 TRNSYS Horizontal Radiation Models	84
6.2 TRNSYS Tilt Radiation Models	84
6.3 Summary of Optics Models	98
6.4 Predicted Components of Tilt Radiation	106
6.5 Collector Parameter Estimates	113
6.6 Parameters for Efficiency and Useful Energy Collector Performance Equations	118
6.7 Parameters for $F_R U_L = 3$ and 3.6	123
6.8 Results with $F_R U_L$ a Linear Function of Wind Speed	123
6.9 Predicted Useful Energy for November 14 - 17, 1996	129
B.1 Typical Property Values	148

List of Figures

2.1 Collectors and Outdoor Weather Station	10
2.2 Drain-Back System Instrumentation Schematic	12
2.3 ICS System Instrumentation Schematic	14
2.4 Natural Convection HX System Instrumentation Schematic	16
3.1 Thermocouple Calibration Curve	26
3.2 Tank Thermocouple Tree	30
3.3 Micro Motion Pressure Drop Curve	31
3.4 Weather Data for Drain-Back and ICS System Tests	35
3.5 Weather Data for Natural Convection Heat Exchanger System Test	35
4.1 Thermocouple Tree in Auxiliary Tank	41
4.2 Auxiliary Tank Temperature Profile	41
4.3 Tank Heat Loss Coefficient Determination	42
4.4 Drain-Back Instrumentation with Heat Trap Installed	46
4.5 ICS Instrumentation with Heat Trap Installed	47
4.6 Natural Convection Heat Exchanger Instrumentation with Heat Trap Installed ..	48
5.1 Weather Station and System Instrumentation	57
5.2 Main Solar Storage Tank Temperatures	59
5.3 Heat Exchanger Inlet and Outlet vs. Time	60
5.4 Heat Exchanger Effectiveness	62
5.5 Log(Ra) vs. Log (Nu)	71
5.6 TRNSYS Tank Model	74
6.1 Beam Refraction	87
6.2 (a) (ta) Angle Relationship; (b) (ta) 1/Angle Relationship	92
6.3 F_R and $F_R U_L$ Variation with Plate Temperature	104

6.4 Boes Horizontal and Hay & Davies Tilt Radiation Models	107
6.5 Erbs Horizontal and Perez Tilt Radiation Models	108
6.6 Reindl Horizontal and Reindl Tilt Models	109
6.7 Diffuse Fraction versus Clearness Index	110
6.8 Predicted Useful Energy ($\eta \cdot A_c G_T$) Over Time for All Parameters Predicted ..	116
6.9 Predicted Useful Energy Using Efficiency and Useful Energy Collector Performance Equations	117
6.10 Instantaneous $F_R U_L$	119
6.11 Instantaneous Parameter Variation	120
6.12 S_{TT} and S_{BD} , $F_R U_L=3$	122
6.13 S_{TT} and S_{BD} , $F_R U_L=3.6$	122
6.14 Wind Speed Variation	124
6.15 $F_R(\tau\alpha)_n$ vs. Percentage of Incident Radiation	128
6.16 Predicted Useful Energy using SRCC Parameters	129
6.17 Predicted Useful Energy, November 14-17, 1996	130

Nomenclature

A	=	Area, m^2
C	=	Capacity Rate, W/C
C_p	=	Specific Heat, J/kgC
D	=	Representative Diameter, m
F	=	Fin Efficiency Factor
F'	=	Collector Efficiency Factor
F_R	=	Collector Heat Removal Factor
G	=	Incident Radiation, W/m^2
Gr	=	Grashof Number
IAM	=	Incident Angle Modifier
L	=	Representative Length, m
LMTD	=	Log Mean Temperature Difference, C
M	=	Number of Data Points
N	=	Number of Nodes in a Tank
Nu	=	Nusselt Number
N_{TU}	=	Number of Transfer Units
P	=	Perimeter, m
Pr	=	Prandtl Number
Q	=	Rate of Energy, W
R	=	Ratio of Tilt Radiation to Horizontal Radiation
Ra	=	Raleigh Number
Re	=	Reynolds Number
S	=	Absorbed Solar Flux, W/m^2
Sc	=	Solar Constant, W/m^2

T	=	Temperature, C
U	=	Overall Heat Loss Coefficient, W/m^2C
$\dot{V}$	=	Volumetric Flow Rate, m^3/s
g	=	Gravity, m/s^2
h	=	Convective Heat Loss Coefficient, W/m^2C
k	=	Thermal Conductivity, W/mC
k_T	=	Clearness Index
$\dot{m}$	=	Mass Flow Rate, kg/s
r	=	Radius, m
t	=	Time, s

Greek Symbols

α	=	Absorptance
α_{sc}	=	Solar Altitude Angle, Degrees
β	=	Angle of Refraction, Degrees
ΔP	=	Pressure Differential, Pa
ΔT	=	Temperature Difference, C
Δz	=	Height Difference, m
ε	=	Effectiveness
γ	=	Collector Tilt Angle, Degrees
η	=	Efficiency
ρ	=	Density, kg/m^3
ρ_g	=	Ground Reflectance
ϕ	=	Relative Humidity
θ	=	Angle of Incidence, Degrees
τ	=	Transmittance
$(\tau\alpha)$	=	Transmittance-Absorptance Product

Subscripts

a, ambient=	Ambient Temperature
ave	= Average
aux	= Auxiliary Tank
b	= Beam Radiation
BD	= Beam and Diffuse Radiation Model
BSDG	= Beam, Sky Diffuse, and Ground Reflected Radiation Model
coll	= Collector
conv	= Convective Flow
diff	= Diffuse (Sky Diffuse and Ground Reflected)
drw	= Draw
gnd	= Ground Reflected
H	= Radiation on the Horizontal
HA	= Hot Inlet on Auxiliary Tank, TRNSYS Model
HM	= Hot Inlet on Main Tank, TRNSYS Model
i	= inside
incident	= Incident Energy on the Collector
in,aux	= Flow of Fluid Into the Auxiliary Tank
in,col	= Temperature of Fluid Into Collector
in,gly	= Temperature of Glycol Into Natural Convection Heat Exchanger
in,HX	= Heat Exchanger Inlet
in,MTnk	= Flow of Fluid Into the Main Storage Tank
in,pmp	= Flow of Fluid Into the Pump
LA	= Replacement Inlet on Auxiliary Tank, TRNSYS Model
LM	= Replacement Inlet on Main Tank, TRNSYS Model
load	= Draw Energy from the System
main	= Main Tank
max	= Maximum
meas	= Measured Value
min	= Minimum

n	=	Normal Incidence
NA	=	Bottom Outlet On Auxiliary Tank, TRNSYS Model
NH	=	Bottom Outlet On Main Tank, TRNSYS Model
o	=	outside
out,col	=	Temperature of Fluid Out of Collector
out,gly	=	Temperature of Glycol Out of Natural Convection Heat Exchanger
out,HX	=	Heat Exchanger Outlet
out,pmp	=	Flow of Fluid Out of the Pump
p	=	Absorber Plate
pmp	=	Pump
sky	=	Sky Diffuse
sup	=	Supply
T	=	Radiation on the Tilt
tnk	=	Tank Average Temperature
TnkBot	=	Tank Bottom Temperature
TnkMax	=	Tank Maximum Temperature
tot	=	Total
TT	=	Total Tilt Radiation Model
useful	=	Useful Energy at the Collector
w,wnd	=	Wind
1A	=	Draw Outlet on Auxiliary Tank, TRNSYS Model
1H	=	Draw Outlet on Main Tank, TRNSYS Model

CHAPTER 1. INTRODUCTION

1.1 Introduction

In the U.S. today there is movement towards being more environmentally aware and friendly, hence there is an increased interest in the use of solar energy, an inherently clean energy. Walter Short of the Solar Energy Research Institute states that 20 years worth of electricity displaced by solar, in the U.S., represents well over 50 tons of carbon dioxide emissions [Rosenbaum, 1991]. As the population continues to increase, fuel prices continue to rise, and the reserves of fossil fuels continue to be depleted, this interest will be intensified.

One use for solar is to heat water for domestic use. The U.S. receives 46,000 quads (quadrillion Btu's) of energy from the sun each year and 1/1000 of that could supply the energy required to produce all the hot water needed for residential and commercial use (about 30 quads) [Bennett, 1995]. The difficulty is that the energy intensity is low (so large areas are needed), the energy is intermittent (some form of storage is needed), and the need for back-up is not eliminated (increases cost). In today's solar market, emphasis is placed on the design of solar systems which exhibit to the consumer a higher ratio of performance to price. In order to improve the performance, systems must be thoroughly understood. This includes both the entire system operation and on a component basis. As research into Solar Domestic Hot Water (SDHW) systems progresses the cost effective-

ness will increase until installing a SDHW system becomes a viable option or a standard feature; some utilities are currently investigating the use of SDHW systems for their customers.

One example of what is possible exists in Sacramento. The Sacramento Municipal Utility District (SMUD) has implemented a program to augment their customers' electric hot water heaters with solar to reduce the summer peak demand. A beneficial coincidence is that a solar hot water system's peak performance occurs with the utilities summer peak demand. The 10 year goal is to replace with solar 34%, or 12,500, electric hot water system systems in the single family residential sector. The average installed system saves 65% of the electricity required for hot water and essentially eliminates electricity demand during summer peak periods. Based on a billing analysis, about 2,331 kWh/yr are saved for a four-person household [Osborn et al., 1995].

The incentive for a customer to install a solar hot water system is a performance related rebate based on projected savings. The rating is obtained through prescribed short-term and component testing under the Solar Ratings and Certification Corporations (SRCC) guidelines [1994]. From the data gathered during testing, long-term performance predictions are made with the use of computer modeling.

For these programs to be successful, now and in the future, SDHW systems need to continue to be technically improved and the SRCC must provide a fair and accurate rating of the systems. This dissertation provides an in-depth analysis of three typical SDHW systems leading to a more thorough understanding based on first principles; thus, leading to technological improvements. The results from testing are compared to the predicted results using the SRCC ratings for the same systems, yielding suggestions for changes to

the ratings procedure. The next section discusses the background in developing a ratings program for SHDW systems.

1.2 Background

The testing and monitoring of Solar Domestic Hot Water (SDHW) systems has been conducted for many years to aid in the design and optimization process and for acceptance testing and certification corroboration. The desirable features of any testing procedure are [Bosanac, 1987]:

- 1) That it is applicable to SDHW systems regardless of design.
- 2) That the system parameters are independent of testing conditions and that prediction is possible for various meteorological and operating conditions as well as certain modifications to the system installation.
- 3) If the tests are used to find a problem, that the results allow a tracing back to the problem source.
- 4) That the test procedure and parameter identification is as simple as possible so as not to impose a significant cost burden.
- 5) That there is sufficient accuracy to permit a meaningful ranking of candidate systems.

The full economic benefit of a solar water heating system cannot be realized unless it is properly designed and sized for its intended application. Needed at this time are design methods which provide reliable estimates of the long-term average annual thermal performance for the various types of systems commercially available [Buckles and Klein, 1980].

For several years, the monitoring of systems over long periods of time was the only method available to assess system performance. The time frame was substantial and a comparison of different data sets was difficult. Recently, short term tests used with computer modeling for long-term predictions have been adopted [Beale, 1987]. But, a fundamental problem is reconciliation of the short-term tests with performance data.

There are many models available to predict the performance of systems. There are models based on the operation of the system as a whole and models based on the operation of each of the components in the system.

One method of whole system testing is followed by the European Solar Collector and Systems testing group. There are no component parameters; a black-box approach is used with input and output parameters determined from all day tests. This testing and modeling allows extrapolation of data beyond the testing conditions [Bourges et al., Part 1, 1991]. No intrusive measurements are necessary, which makes it possible to monitor systems already installed without any interruption to their operation [Arkar et al., 1991]. Three system parameters are determined which can be attributed to the gains, losses and threshold effects of the system. Results of analysis of simulated data imply that significant scatter arises from the input-output model itself [Bourges et al., Part II, 1991]. This model is not well suited for design and optimization of solar hot water systems [Schreitmuller, 1987].

A similar model is a dynamic testing method of solar system testing developed at L-M University in Munich. This model also takes the black-box approach, but there is a time dependence in the model, as opposed to stationary models. The time dependency

accounts for any load and radiation profiles and any time sequence can contribute to system parameter identification [Spirkl, 1987].

The second approach, the modeling of the components, is used by a model developed at the University of Wisconsin called TRNSYS (TRaNsient SYstem Simulation) [Klein et al., 1994]. There are several others available, such as SHOW (Solar Hot Water) which will simulate one and two tank thermosyphon and pumped flat-plate SDHW systems [Baughn and Young, 1984], but TRNSYS is the most widely used. TRNSYS is the modeling program used by the SRCC for system rating. Each component is modeled separately in TRNSYS, then a system model is "pieced" together from component models. A study by Fanney and Liu [1980] compared the results of two years worth of testing on six SDHW systems with annual TRNSYS predictions. There was agreement to within 11%. Although this estimate of annual performance is reasonable, predictions of short term performance may not be as good because of the complicated interactions between components. In particular, even though the predicted useful energy gained at the collector is reasonably accurate, the predicted collector outlet temperature is incorrect.

Further background is provided in each of the chapters as pertaining specifically to the particular research being presented.

1.3 Objective

The performance rating of a SDHW system is important in assessing cost savings and in optimizing the system. Therefore the testing procedures and the modeling are equally important. The current test procedures require very specific component testing

which may not capture the component interactions such as the effect of storage stratification on the collector efficiency.

This dissertation presents the results of detailed in-situ tests of three SDHW systems. Various tests, such as normal operation tests, are performed on the systems which provide information on the interaction of the components. Also tests are performed which yield component information, but the test data from these tests are not independent of the rest of the system, such as heat loss tests on the storage tanks.

The results from these comprehensive tests are used to analyze all of the systems and their components. This analysis aids in the verification of, and improvements to, the ratings procedure for domestic hot water systems by lending confidence to the current models, by improving models, or by suggesting alternative procedures to follow to gain additional performance parameters. The understanding gained by the in-depth analyses of the systems suggests design improvements to the systems.

1.4 Organization of Dissertation

The current state of research in solar domestic hot water systems is discussed in chapter one. Descriptions of the systems tested, and their unique features, are provided in chapter two. The test protocol is also presented in chapter two.

Energy balances, which help determine the necessary instrumentation, are presented in chapter three. Also, the instrumentation of the systems is described in chapter three.

Strategically selected components are then analyzed individually. The analysis of the indoor hot water storage tanks used for each of the systems is covered in chapter four.

Two of the systems tested use specially designed heat exchangers to transfer the collected solar energy to the load water; the heat exchanger for one of these systems is analyzed in chapter four. An in-depth analysis of the most important feature of the systems, the flat-plate collectors, is presented in chapter six.

The conclusions and recommendations for further work are presented in chapter seven.

CHAPTER 2. DESCRIPTION OF SYSTEMS AND TESTS

2.1 Introduction

Three solar domestic hot water heating systems that are presently on the market have been installed and instrumented in detail. The systems each have at least one unique feature and are being tested simultaneously to assess similarities and differences in performance. A description of the systems tested is given in this chapter. Also presented are the testing procedures followed throughout the testing period.

2.2 Description of Systems

The three systems include one of each type of typical system currently on the market. These are: (1) a relatively low-cost unpressurized drain-back system with a load side heat exchanger, (2) an integral collector storage (ICS) system which uses a pump to circulate water between the solar storage (collectors) and the auxiliary storage tank, and (3) an ultra-low collector flow rate system, utilizing glycol in the collector loop, with a natural convection heat exchanger used to transfer the collected heat to the water in the solar storage tank (natural convection heat exchanger system). Table 2.1 provides the system characteristics.

Table 2.1 System Details

System Characteristic	Drain-Back	ICS	Nat. Conv. HX
Net Collector Area, m ² (ft ²)	5.52 (59.4)	2.6 (28) ^a	5.67 (61.0)
Solar Tank Volume, m ³ (gal.)	0.35 (91.4)	-	0.21 (56.4)
Aux. Tank Volume, m ³ (gal.)	0.14 (36)	0.14 (36)	0.14 (36)
Collector Flow Rate, m ³ /s (ft ³ /s)	5.7×10^{-5} (2.0×10^{-3})	5.9×10^{-5} (2.1×10^{-3})	1.8×10^{-5} (6.4×10^{-4})
Pump Motor Size, W (hp)	49.7 (1/15)	7.5 (1/100)	248.6 (1/3)
Controller DT - On/Off, °C	7/2	4.5/2	7/2

a. Aperture area includes gaps between tubes [Smith, 1995]

2.2.1 Overview of Systems

The collectors are mounted facing due south at a 45° slope on the collector test bed located about 6.1 m (20 ft.) south of the basement of Solar House 1 at Colorado State University. From there, piping is run into the basement of Solar House I, where all storage tanks (except the ICS collectors), pumps, and the external heat exchanger are located. Also in the basement of Solar House I is the HP system 10 data acquisition system and the 80386 computer used for data logging.

The plumbing between the basement and the test bed is located outside, except for 2.4 m (8 ft.), which is located inside a small annex used to house a dessicant wheel. All systems are connected in parallel to both the water main supply line and the hot water draw line. Except for the 0.0127 m (1/2 in.) sweated copper piping near the pumps, all pip-

ing is 0.0187 m (3/4 in.) sweated copper. All lines are well insulated. To minimize pressure drop, all valves are 0.0187 m (3/4 in.) or 0.0127 m (1/2 in.) through-flow ball valves.

2.2.2 Ambient Weather Station

The weather station, shown in Fig. 2.1, consists of two pyranometers, used to measure horizontal and tilt radiation, a wind speed anemometer with a direction indicator, and an outdoor ambient temperature sensor.

Figure 2.1 Collectors and Outdoor Weather Station

2.2.3 Drain-Back System

The drain-back system consists of three panels connected in parallel, with an overall outside dimension of 4.88 m x 1.18 m (196 in. x 46.5 in.). The net aperture area of the collectors is 5.9 m² (65 ft²). The collector uses an ethylene propylene dyene monomer (EPDM) absorber, which is a pliable plastic not subject to damage by freezing. There is about 292.6 m (960 ft.) of black EPDM tubing, 0.0089 m (0.35 in.) outside diameter and 0.0057 m (0.225 in.) inside diameter, encased in the collectors as an absorber. The absorptance of the EPDM is 0.9. The single cover is 0.0064 m (0.25 in.) Lexan honeycomb. The cover is designed to bow due to thermal expansion (similar to a bimetallic strip) creating a vent to ambient, thereby protecting against high stagnation temperatures. The frame is manufactured from sheet aluminum.

A schematic of the drain-back system is shown in Fig. 2.2. The return flow from the drain-back collector enters a custom 0.35 m³ (91 gallon) unpressurized solar storage tank. The pump circulates the solar storage tank water to the collectors and back to the tank. Heat is removed from the solar storage tank using a load-side heat exchanger. The heat exchanger, which spans nearly the entire height of the tank, is 36.58 m² (120 ft²) of 0.019 m (0.75 in.) copper tubing coiled inside the solar storage tank. During a draw potable water from main supply flows through the heat exchanger coil located in the solar storage tank and then into the bottom of a standard 0.14 m³ (36 gallon) auxiliary storage tank. Energy is transferred between the tanks only when there is a hot water draw.

The pump is operated by a differential controller. The upper dead band is set at 7°C, and the lower dead band is set at 2°C. The thermistors used for measuring the control temperatures are located on the outlet of the collector and on the outside of the bottom of

Figure 2.2 Drain-Back System Instrumentation Schematic

the solar storage tank. The pump is an impeller type rated at 49.7 W (1/15 hp) that circulates water at a rate of $5.7 \times 10^{-5} \text{ m}^3/\text{s}$ (0.9 gpm).

2.2.4 Integral Collector Storage System

The ICS system consists of two collector modules, each with four evacuated tubes. Each evacuated tube contains 0.019 m^3 (5 gallons) of integral storage. The inner tube which holds the water is coated with a selective surface of black chrome with an absorptance of 0.93. The net aperture area (projected absorber area) is 1.82 m^2 (20 ft^2). The projected area of the modules is 3.32 m^2 (36 ft^2). The two modules were initially connected in series, and later in parallel, for the tests. The heat loss coefficient of these collectors is sufficiently low such that they are able to maintain temperatures above freezing for several days of inclement weather and no additional freeze protection is required. However, the pipes to and from the collector are subject to freezing, and consequently, rupture. They are drained during periods of freezing weather.

The ICS system plumbing is shown in Fig. 2.3. An induction pump, rated at 7.46 W (1/100 hp) circulates the water between the collector storage and auxiliary tank at a rate of $5.9 \times 10^{-5} \text{ m}^3/\text{s}$ (0.94 gpm). The pump was initially installed to prevent overheating of the water in the collectors. During a draw, the pump is inactivated due to the water mains supply pressure, and water mains supply flows out to the collectors. Since a draw is about 0.079 m^3 (21 gallons), essentially the water from one collector module is displaced with main supply water. The displaced collector water flows into the bottom of a standard auxiliary tank. During a draw, energy is removed from the top of the auxiliary tank.

Figure 2.3 ICS System Instrumentation Schematic

The pump is operated with a differential controller with the upper dead band set at 4.5°C and the lower dead band set at 2°C. The thermistors that measure the temperature difference are located on the outlet of the collector and on the bottom of the auxiliary storage tank.

2.2.5 Natural Convection Heat Exchanger System

The natural convection heat exchanger system consists of two standard 1.2 m x 2.4 m (4 ft. x 8 ft.) micro-flow collectors connected in series. The solar absorber consists of two aluminum fins metallurgically bonded to a copper tube and constructed in a serpentine fashion. The absorber plate is an anodic cobalt selective surface with an absorptance of 0.93. The cover is manufactured of low-iron, tempered glass. The supply tubing to the collector is 0.0079 m (5/16 in.) diameter and the return tubing is 0.0064 m (1/4 in.) diameter. There is 7.62 m (25 ft.) of each.

A system schematic is presented in Fig. 2.4. This system utilizes a glycol fluid pumped between the collectors and the heat exchanger at a rate of $1.8 \times 10^{-5} \text{ m}^3/\text{s}$ (0.3 gpm). The glycol helps to protect against freezing. The pump is an impeller type rated at 248.7 W (1/3 hp), and is operated by a differential controller. The thermistors that measure the controlling temperature difference are located on the outlet of the collectors and inside the heat exchanger shell. The upper dead band is set at 7°C and the lower dead band is set at 2°C.

Upon returning from the collectors, the glycol enters the coil of a high performance heat exchanger, located adjacent to the solar storage tank. The heat exchanger is a shell and coil type with four copper coils enclosed in a copper shell. There is 30.48 m (100

Figure 2.4 Natural Convection HX System Instrumentation Schematic

ft.) of 0.0064 m (1/4 in.) diameter tubing, with a heat exchanger area of 0.61 m² (6.5 ft²). This tubing constitutes four concentric coils for an overall height of 0.41 m (16 in.). The glycol flows simultaneously through each coil. This, together with the small diameter tubing to and from the collector, accounts for the pump being much larger than for the other two systems. The overall heat transfer coefficient is 380 W/m²K (67 Btu/hr ft² °F).

Water, inside the shell, circulates from the heat exchanger to the solar storage tank via natural convection, driven by the temperature difference between the hot glycol and the colder solar storage tank water. During a draw, mains supply flows into the bottom of the solar storage tank while the hot water at the top of the solar storage tank is transferred to the bottom of a 0.14 m³ (40 gallon) auxiliary tank. Energy is removed from the top of the auxiliary tank.

2.3 Test Procedures

Three separate tests are performed on each of the systems. Attempts were made to test the systems simultaneously. This was an attractive plan, but it did not work out well because the natural convection heat exchanger system solar storage tank was missing a part that affects the stratification of the tank. Therefore, the testing on the natural convection heat exchanger system is accomplished separately from the other two systems.

The three tests performed are: (1) normal operation tests, which are used to determine the overall system performance, (2) continuous operation tests, which aid in determining collector parameters, and (3) heat loss tests on the tanks, which are used to determine the heat loss coefficients of the tanks. The data are taken every 15 seconds for the duration of each test. The data acquisition system monitors 88 channels. To reduce the

data to a manageable set, five minute averages are used, except for the draw data. Since a draw occurs over only seven minutes, the 15 second data are used to compute the energy withdrawn from the tank during this time. The 15 second data are reduced using the program included on a disk as Appendix C.

2.3.1 Normal Operation

The normal operation tests are performed so as to compare results to the SRCC ratings, given in OG-300, a directory of solar water heating systems meeting minimum operating and performance requirements [1993], for these systems. The SRCC ratings are based on TRNSYS [Klein et al., 1994] runs with standard decks. The decks are based on ASHRAE standard 95-1987 test procedures [1988]. The present systems are tested in accordance with ASHRAE procedures. The systems are allowed to operate normally for a period of five days with draws being performed three times a day, at 8:00 a.m., Noon, and 5 p.m. The draw flow rate is $2 \times 10^{-4} \text{ m}^3/\text{s}$ (3 gpm) for seven minutes, or approximately 0.08 m^3 (21 gallons) per system per draw. Before beginning a test, a manual draw is performed on all the systems, either before sunrise or after sunset, until a constant temperature is monitored throughout the test set-up. If the outside ambient temperature is below freezing, tests are not conducted since the ICS system does not have freeze protection for the piping running outside, which are drained down during these periods.

2.3.2 Continuous Operation

For the continuous operation tests, the systems are forced to operate unnaturally. The pumps are manually turned on, and left on throughout the course of the three day test. These tests are used in the analysis to determine the collector parameters such as the collector loss coefficient ($F_R U_L$), and the absorber transmittance/absorptance product ($F_R(\tau\alpha)$). These tests are performed for three configurations of the systems: 1) no draws are performed and no auxiliary heat is used, 2) draws are performed, and no auxiliary heat is used, 3) draws are performed, and auxiliary heat is used. Again, the system temperatures are brought to equilibrium before beginning the test.

2.3.3 Heat Loss Tests

These tests are performed on the tanks alone, to determine the heat loss coefficients. Here, the tanks are filled with constant temperature hot water (at least 10°C hotter than the ambient room temperature) and allowed to sit, while being monitored, for at least 12 hours. The temperature decay over time is monitored. The valves on the piping to/from the tanks are positioned opened or closed, depending on the test. More detail is given in Chapter 4.

2.4 Summary

This chapter has presented the details of the systems and the philosophy of testing. The next chapter presents the instrumentation and the experimental details.

CHAPTER 3. EXPERIMENTAL APPROACH AND TESTING METHODOLOGY

3.1 Introduction

The final instrumentation for the experiments, the calibration procedures used, and an enumeration of the experimental revisions made during the course of the testing are presented and discussed in this chapter.

It is desired to make sufficient measurements to enable detailed energy balances to be performed on all significant components of the systems, and by extension, on the entire systems in aggregate. To accomplish this, it is necessary to make detailed measurements on the systems themselves. Significant instrumentation is required to be able to identify the transient nature of the systems' operation.

A comprehensive instrumentation plan of the highest quality is designed to effect system measurements of temperatures, electrical power input, flow rate, and differential pressure across the pumps. In addition, an outdoor ambient weather station has been designed and installed specifically for these tests. Measurements from the ambient weather station constitute inputs while system measurements constitute system states and outputs. The goal of the design is to be able to obtain physical energy balances on all major components to within 10% of delivered energy. The density of the instrumentation is such as to enable detailed component-by-component analysis, as well as whole system analysis. Finally, there is sufficient redundancy via independent measurements to permit

the determination of accuracy experimentally. Such measurement detail meets the dual objectives of analysis: (1) correlation of inputs with outputs, and (2) detailed system and component analyses.

The instrumentation evolved over time to meet specific needs identified during the course of the work. After initial data acquisition runs were made and subsequent analyses performed, the instrumentation was revised many times over the course this work. A complete instrumentation list is given in Appendix A.

3.2 Energy Balances

In order to perform the detailed component-by-component analyses, and to meet the overall system energy balances goal of 10%, the following types of energy balances need to be accommodated by the instrumentation system.

3.2.1 Convective Energy

The energy gain/loss across all major system components, including collectors, heat exchangers, storage tanks, and long piping runs (from the indoor storage tanks to the collectors) needs to be determined during times when there is fluid flow. The appropriate equation is given as:

$$\dot{Q}_{convective} = \rho C_p A \dot{V} \Delta T \quad (3.1)$$

The volume flow rates and the temperature differences are measured directly, the thermodynamic variables are determined from the thermodynamic state, and the area of the pipe is determined from the piping size.

Note that measurements of this type are taken across both sides of the heat exchangers, enabling the losses to be determined. The heat exchanger effectiveness, an important design parameter for active solar systems, is also determined.

3.2.2 Parasitic Power

The parasitic power consumed by the fluid being pumped is calculated as:

$$\dot{E}_{fluid} = \dot{V}(\Delta P + \Delta z g) \quad (3.2)$$

Both $\dot{V}$ and ΔP are measured. The electrical power is also measured, therefore the overall efficiency (combined efficiency of the pump and the motor) is obtained by calculation, as follows:

$$\eta_{overall} = \eta_{motor} \eta_{pump} = \frac{\dot{E}_{fluid}}{\dot{E}_{electrical}} \quad (3.3)$$

3.2.3 Internal Energy

In order to determine the heat loss coefficients for the tanks, the internal energy of the water in the tank is calculated as the time rate of change of the temperature of the mass of water in the tank. This internal energy is then used to determine the heat loss coefficient of the tank, as presented in Chapter 4. The equation used is:

$$\dot{Q}_{\text{tank}} = MC_p \frac{d\Delta T}{dt}, \quad \Delta T = T_{\text{tank}} - T_{\text{ambient}} \quad (3.4)$$

The thermocouple arrays in the tanks have been designed to yield vertical temperature profiles throughout the tank. These temperatures are monitored over the duration of a heat loss test. The mass of the water in the tank was measured on a precision balance scale that had been calibrated two weeks earlier.

3.2.4 Overall System Energy

An overall energy balance is performed on each of the systems. To include as many measured terms as possible, energy balances are performed around both the solar storage tanks and around the auxiliary tanks. The collector is treated separately in Chapter 6. Since each system is unique, so is the resulting energy balance equation. The ΔT 's used in the following equations are to account for the energy due to tank heat losses; each ΔT is the volume weighted temperature of the respective tank, minus the room ambient temperature. The energy balances for the drain-back system are:

$$\begin{aligned} MC_p \frac{dT}{dt}_{\text{Main}} &= \dot{m}_{\text{coll}} C_p (T_{\text{in, MTnk}} - T_{\text{in, pmp}}) - \dot{m}_{\text{drw}} C_p (T_{\text{in, aux}} - T_{\text{sup}}) - UA\Delta T_{\text{Main}} \\ MC_p \frac{dT}{dt}_{\text{Aux}} &= \dot{m}_{\text{drw}} C_p (T_{\text{drw}} - T_{\text{in, aux}}) - UA\Delta T_{\text{Aux}} + Q_{\text{Aux}} \end{aligned} \quad (3.5)$$

The energy balance equation for the ICS system is:

$$\begin{aligned} MC_p \frac{dT}{dt}_{\text{Aux}} &= \dot{m}_{\text{pmp}} C_p (T_{\text{in, MTnk}} - T_{\text{in, pmp}}) - \dot{m}_{\text{drw}} C_p (T_{\text{drw}} - T_{\text{in, MTnk}}) \\ &\quad - UA\Delta T_{\text{Aux}} + Q_{\text{Aux}} \end{aligned} \quad (3.6)$$

Finally, the energy balance equation for the natural convection heat exchanger system is:

$$\begin{aligned}
 MC_p \frac{dT}{dt}_{Main} &= \dot{m}_{conv} C_p (T_{out, HX} - T_{in, HX}) - \dot{m}_{drw} C_p (T_{in, aux} - T_{sup}) - UA \Delta T_{Main} \\
 MC_p \frac{dT}{dt}_{Aux} &= -\dot{m}_{drw} C_p (T_{drw} - T_{in, aux}) - UA \Delta T_{Aux} + Q_{Aux}
 \end{aligned} \tag{3.7}$$

3.3 Instrumentation

Outdoor and system measurements are made as indicated in the Figures 2.1 - 2.4. The types of measurements include: temperatures, fluid flow rates, electrical power, and pressure differential across the pumps. Measurements are taken every 15 seconds and averaged to yield 5-minute data. For analysis specific to draw data (e.g. energy drawn from the tank) the 15-second data are used. The following descriptions will be clearer by referring to the figures.

3.3.1 Calibrations

Except for the differential pressure sensors and the Coriolis flow meter, which are new and calibrated by the manufacturer, all instruments were calibrated through the Data Acquisition System (DAS) used to measure and store all of the experimental data. It is critical that the calibration be made through the same DAS with which the measurements are taken, so as to calibrate the instrument/DAS system rather than just the sensors alone. This ensures both accuracy and correctness (for example, that the instruments are connected properly to the DAS).

Two Eppley PSP pyranometers are used to measure the incident solar radiation; one measures the horizontal radiation, the other faces south and measures the 45° tilt radiation. The pyranometers were calibrated versus a known pyranometer at the Colorado State University Atmospheric Sciences Laboratory in the summer of 1993. Each pyranometer was mounted in the orientation used for data collection (horizontal or tilted). The calibration test proceeded over the course of eight days. The least squares fit for the horizontal pyranometer yielded a calibration constant of $1.063813 \times 10^{-5} \text{ V/W/m}^2$ with a standard deviation of $4.352 \times 10^{-7} \text{ V/W/m}^2$ and an r^2 of 1.0. The least squared fit for the tilt pyranometer yielded a calibration constant of $9.963726 \times 10^{-6} \text{ V/W/m}^2$ with a standard deviation of $4.3259 \times 10^{-7} \text{ V/W/m}^2$ and an r^2 of 1.0.

The type T thermocouples were calibrated versus a precision mercury thermometer, readable to 0.05 °C. Six thermocouples were calibrated - four from one roll of wire, and two from a second roll of wire (from both the beginning and ends of the rolls). Absolute temperature errors of less than 0.05 °C were observed. The calibration curve is shown in Fig. 3.3, with a standard deviation in absolute temperature of less than 0.02 °C. Based upon these observations, it is concluded that no additional thermocouples need be calibrated [Davidson, 1993]. An accuracy of $\pm 0.1^\circ\text{C}$ will be used for the error analysis to account for the unknown placement of the thermocouple in the flow stream profile.

All of the turbine flow meters were calibrated using a manometer and an ASME calibrated venturi. In regards to orientation and entrance length necessary for fully developed flow to be measured, the flow meters were installed in the system and in the calibration stand according to the guidelines given by the manufacturer [EG & G, 1990] and in

Figure 3.1 Thermocouple Calibration Curve

the literature [Shafer, 1961]. Such devices directly yield the volume flow rate. An equation of state is required to determine the mass flow rate.

Six Hall effect AC Watt transducers are used for power determination. Three large transducers, rated at 20 kW output with external coil current transformers, are used on the auxiliary heating elements, and three transducers rated at 120 W output are used on the pumps. One of the large Hall effect transducers was sent back to the factory in the summer of 1993 for factory calibration, and the others were calibrated against this device. The smaller transducers were calibrated per the instructions given by the manufacturer of the large transducers [Ohio Semitronics, 1993].

Table 3.2 presents the types of instrumentation used and their associated errors. A complete error analysis is provided as Appendix B.

Table 3.1 Instrumentation

Instrumentation	Characteristic	Comments	Error
Thermocouple	Type T	High Grade	+/- 0.1 °C
Flow Meter	Turbine, Vol.	Drain-Back, HX Collector, Supply ICS, Drain	+/- 0.4% +/- 0.5%
	Coriolis, Mass	HX Solar Storage	+/- 0.2%
Pressure Xducers	Differential	Across the Pumps	+/- 1% full scale
Pyranometers	PSP	Horizontal, Tilt	+/- 2%
Watt Xducers	Hall Effect	On Pumps, Controllers	+/- 0.5%

3.3.2 Ambient Weather Station

Measurements taken are: solar irradiation (total horizontal and total tilt on the collector plane), ambient dry-bulb temperature, and wind speed and direction. The ambient temperature sensor is shielded from direct and reflected solar radiation and long wave sky radiation by a double-walled aspirated housing that is painted white [European Solar Collector and Systems Testing Group, 1989; ASHRAE, 1988]. Tests have shown this type of temperature sensor to be accurate to within 0.2 °C, even in conditions of high solar radiation. The ambient dry-bulb temperature and wind measurements are taken at a height corresponding to mid height of the collector [ASHRAE, 1988], adjacent to the collectors. Thus, it may be possible to correlate the overall collector loss coefficient with the convective loss coefficient (h) due to wind speed and direction, i.e. h as a function of the Reynolds number, Prandtl number, and collector geometry.

3.3.3 Temperature Measurements

All temperature measurements are made of high-grade type T (Copper/Constantin) thermocouple wire. It is essential that extremely accurate temperature measurements be made, as the temperature differences are directly proportional to the energy balances on components. All thermocouples are shielded using electrical conduit, and electrically isolated from the shields to prevent ground loops. All temperatures are measured absolutely. Temperature measurements in flow streams (except for the small tubing on the Thermo Dynamics collector loop) are made using identical fittings, with the result that any systematic errors due to geometry will cancel [McGee, 1988]. Finally, where there are junctions involving dissimilar metals, a thermal heat sink is provided and the junctions are insulated so as to minimize spurious EMF's due to non-isothermality [ASTM, 1981].

For fluid temperature measurements, thin-walled brass tubing and Swagelock compression fittings are inserted in 0.0254 m (1 in.) wyes to isolate physically the thermocouple from the fluid. They are installed no more than 200 mm (8 in.) from the inlet/outlet of the point of interest, and the probe is directed upstream [European Solar Collector and Systems Testing Group, 1989]. Since the Biot Number for the brass tube is less than 0.003, the radial resistance can be considered to be negligible. These brass fittings have been designed so that axial conduction along the tube is negligible, and tests have been conducted to demonstrate that this is indeed so. To obtain the significance of the axial conduction, the following equation is solved theoretically:

$$\begin{aligned}
hA(T_{fluid} - T_{brass}) &= kA \frac{dT_{brass}}{dx} \\
x &= \text{length along brass tube} \\
T_{fluid} &= \text{linear in } x \\
@x = l, \quad dT/dx &= 0
\end{aligned}
\tag{3.8}$$

The resulting temperature distribution of the brass tube is an exponential decay, given as:

$$\begin{aligned}
m &= \frac{h}{k} \frac{2r_o}{(r_o^2 - r_i^2)} \\
T_{brass} &= lx + \frac{e^{-mx}}{me^{-ml}}
\end{aligned}
\tag{3.9}$$

The ratio 'm' is at least 300. Therefore, axial conduction is negligible.

In the past, tank temperature measurements have been made by cutting a large hole in the top of the tank and inserting a thermocouple tree. This is not a satisfactory procedure since cutting through the lining of the tanks causes the tank to commence rusting rapidly. Here, thermocouple arrays have been designed to measure tank temperatures at eight heights, where the spacing between thermocouples in the tank is uniform at one-eighth of the height of the tank. Each thermocouple in the array is installed in a thin-walled brass tube, as shown in Fig. 3.4. Plexiglass spacers are provided to align the tubes horizontally. Before soldering on the end caps, thermal paste is applied to the thermocouple junction, which has been wrapped in heat shrink tubing so as to isolate it electrically from the shield. The brass plug holding the thermocouple array is threaded into the 0.019 m (3/4

in.) NPT fitting on the top of the storage tank, normally used for the pressure/temperature relief valve. The pressure/temperature relief valve is “teed” off of the hot water draw line.

Figure 3.2 Tank Thermocouple Tree

3.3.4 Fluid Flow Rate

For most of the fluid flow rate measurements turbine flow meters are used. Turbine flow meters are installed in the draw line, supply line, and in each of the systems to measure the flow out to the collectors.

The hot water draws are performed automatically using solenoid valves connected to a timer. The draws on the systems are performed sequentially. The draw flow rate is nominally $2 \times 10^{-4} \text{ m}^3/\text{s}$ (3 gpm), and continues in each system for 7 minutes for a total of 0.08 m^3 (21 gallons). These draw flows are accurately measured using turbine flow meters.

However, for the very low flow rate on the natural convection side (solar storage tank) of the natural convection heat exchanger, a highly accurate mass flow rate meter manufactured by Micro Motion Corporation is used. This meter acts on the principle of sensing the Coriolis acceleration of the fluid through a vibrating U-tube, thus it is the least invasive with the lowest pressure drop (shown in Fig. 3.3) of all mechanical flow metering devices, making it ideally suited to our application. At the low flow rate seen in the natural convection loop of the heat exchanger ($< 1 \times 10^{-5} \text{ m}^3/\text{s}$), the pressure drop is less than 20 Pascals.

Figure 3.3 Micro Motion Pressure Drop Curve

3.3.5 Differential Pressure Measurements

Differential pressures are measured across the pumps to yield, together with the volumetric flow rate, the parasitic pumping power into the fluid. These strain gage devices, which are very sensitive to overpressurization, were bought new for this project. The transducers are installed using 0.0032 m (1/8 in.) pressure taps soldered to the pipes on either side of the pump. The transducers are connected to the taps via plastic tubing. Valves are installed in the plastic tubing on either side of the tap to control the pressurizing of the transducers. The pressure transducers' measurements can be checked against the electrical pumping power and the fluid volume flow rate for consistency, by assessing whether the overall efficiencies of the pumps are reasonable.

3.3.6 Electrical Power Measurements

Hall-effect transducers are used to measure the power input to the electrical auxiliary heaters in the auxiliary storage tanks, and the collector fluid circulating pumps. Watt-hour meters are also used to measure the electrical input to the auxiliary heaters. These are used as a gross double check, and also to determine at a glance whether the heaters are on. The auxiliary heater usage is measured with large 20 kW rated output transducers with external coil current transformers. The pump power usage is measured with 120 W rated output transducers.

3.4 Overall Energy Balance Results

The results for the system energy balances, as described in section 3.2.4, are presented in Table 3.2. The calculated/measured values of the individual terms in the energy balance equations, (eqn's 3.5 to 3.7) are also shown. Also included in the table is the useful energy gained from the collector which is $\dot{m}_{coll}C_p(T_{out,col}-T_{in,col})$ for flat-plate collector systems (the drain-back and natural convection heat exchanger systems) and is $\dot{m}_{pmp}C_p(T_{out,col}-T_{in,col}) + \dot{m}_{drw}C_p(T_{out,col}-T_{in,col})$ for the ICS system. The incident radiation is calculated as the measured radiation incident on the tilted surface times the net collector area given in Table 2.1. The results for the drain-back and ICS systems are given for the six day testing period from 4 p.m. August 24 to 8:30 p.m., August 30, 1995. The results for the natural convection heat exchanger system are for the eight day testing period from 7 p.m. February 18 to 7 a.m. February 27, 1996. Figures 3.1 and 3.2 present the weather data for these two periods.

The last three rows of Table 3.2 show the system energy balance percentages. The energy balance percentages are based on the energy gained in the tank/tanks. "Main" refers to an energy balance with the control volume around just the main solar storage tank, "Aux" is an energy balance with the control volume around the auxiliary storage tank, and "Overall" is an energy balance with the control volume around both storage tanks. All of the energy balances are within the 10% goal. A large portion of the error may be attributed to the behavior of the storage tanks, which is discussed in Chapter 4. The storage tank heat losses account for 10% to 15% of the overall energy gained in the tanks.

Table 3.2 Energy Balance Results

(MJ)	Drain-Back	ICS	Nat. Conv. HX
Q_{incident}	779.2	395.3	428.1
Q_{useful}	256	219.7 (184.8 due to draws)	281
$MC_p(dT/dt)$ (Main)	5.8	-	8.2
$MC_p(dT/dt)$ (Aux)	0.3	2.2	30.2
$m_{\text{coll}} C_p(T_{\text{in,MTnk}} - T_{\text{in,pmp}})$	231	-	253.9 (glycol at HX)
$m_{\text{pmp}} C_p(T_{\text{in,MTnk}} - T_{\text{in,pmp}})$	-	18.5	-
$m_{\text{conv}} C_p(T_{\text{out,HX}} - T_{\text{in,HX}})$	-	-	247.3
$m_{\text{drw}} C_p(T_{\text{in,aux}} - T_{\text{sup}})$	203.3	-	251.9
$m_{\text{drw}} C_p(T_{\text{drw}} - T_{\text{in,aux}})$	26.5	16.93	163
Q_{load}	229.9	216	414.8
$UA\Delta T_{\text{Main}}$	35.8	-	5.8
$UA\Delta T_{\text{Aux}}$	28.9	37.18	50.74
Q_{Aux}	57.4	42.32	256.2
$Q_{\text{pump, fluid}}$	3.02	0.03	0.59
$Q_{\text{pump, electrical}}$	16.38	0.73	11.75
Energy Balance: Main	6.0%	-	7.5%
Energy Balance: Aux	2.8%	7.0%	7.7%
Energy Balance: Overall	5.4%	7.0%	1.2%

Figure 3.4 Weather Data for Drain-Back and ICS System Tests

Figure 3.5 Weather Data for Natural Convection Heat Exchanger System Test

3.5 Test Revisions

As mentioned in the introduction to this chapter, many changes were deemed necessary to the experimental set-up; either for ease of testing, equipment failure, or as a result of preliminary analyses. Some changes include:

(1) Several thermocouples were added to allow for a more complete energy balance analysis.

(2) The Micro Motion flow meter (located in the thermosyphon loop on the natural convection heat exchanger system) was replaced with a Micro Motion flow meter that has 1/10 the fluid pressure drop.

(3) A tempering valve was added to the draw flow to allow for "energy" draws.

(4) Solenoid valves were added to the draw lines for each system to automate the draws. The output from the solenoid valves were eventually changed to DC outputs once it was discovered that the AC readings were unable to be measured using the data acquisition system.

(5) The turbine flow meter on the ICS system was moved from downstream of the pump to the line going directly to the collector. This makes the determination of the energy gained across the collector easier to calculate since this meter will now also measure the draw flow going out to the collector.

(6) Heat traps were added to the inlet/outlet pipes on the storage tanks. This is discussed in chapter three.

(7) The collector cover on the drain-back system was replaced. The underside of the original cover had become covered with brown residue from outgassing of the EPDM material.

(8) Two thermistors used for the temperature measurements on the pump controller had to be replaced. One was on the natural convection heat exchanger system; the other was on the ICS system.

(9) The frequency to voltage converter chip used with the wind speed anemometer was replaced three times. It is highly sensitive to voltage spikes, and would sometimes fail during lightning storms. Each time the chip was replaced, the anemometer would be recalibrated.

(10) The wind direction vane was ripped apart during a storm and had to be replaced.

(11) The turbine flow meters are old, and many times the electronics failed. The problem was always found and corrected (chips, resistors, capacitors, etc. had to be replaced). The flow meters were always recalibrated when repairs were performed. The turbines on two separate flow meters had to be repaired; they would not spin freely. One was taken apart and cleaned thoroughly, the other had disengaged from its mounting shaft.

(12) The pumps on both the drain-back system and the ICS system were replaced. The ICS system pump failed, and the drain-back system pump was replaced with a higher capacity pump to provide a higher flow rate out to the collectors.

(13) Attempts were made to test the systems simultaneously. This was an attractive approach that did not work out well. After an in-depth analysis of the natural convection heat exchanger system data, it was discovered that the "dip-tube" was missing on the main solar storage tank. The dip-tube directs the mains supply water flowing into the primary solar storage tank to the bottom of the tank. Without the dip-tube, when a draw is performed the supply water will enter the top of the solar storage tank, mix with the existing

top of the solar storage tank, and then enter the auxiliary storage tank. Thus, most of the solar heated water in the solar storage tank remains, and is not delivered to the load.

Unfortunately, this was not discovered until very late in the experiments; therefore, the natural convection heat exchanger system tests were performed by themselves after a new dip-tube was inserted. Fortunately, though, it does not affect the analysis of the components of the system.

(14) When additional channels were added to the data acquisition system, the data acquisition period had to be increased from 8 seconds to 15 seconds.

3.6 Summary

The instrumentation was designed, calibrated, and installed with the purpose of being able to make comprehensive measurements of the solar domestic hot water systems overall and on a component basis. The degree to which energy balances can be made, less than 7%, attests to the successful implementation of the instrumentation plan.

The drain-back system has an overall average pump efficiency of 18%, with a maximum of 20%. The overall average pump efficiency for the ICS system is 4%, with a maximum of 13%, and the overall average pump efficiency of the natural convection system is 5% with a maximum of 6%. If the useful energy at the collector is used as a basis, the electrical pump energy consumes less than 5% of the overall for all three systems. The portion of this energy gained by the fluid is at most 1%. Solar water heating systems have been notorious for inefficient impellers and electrical motors, but overall the energy consumed is a small fraction of the total, so the pumps will not be further analyzed in this work.

CHAPTER 4. STORAGE TANK HEAT LOSS TESTS

4.1 Introduction

The heat loss test performed on the storage tanks and the analyses of the heat loss tests are described in this chapter.

The primary and auxiliary storage tanks of a solar system play an important role in the overall performance of the system. The heat loss from the tanks is typically 10% to 15% of the useful energy gained at the collector. The tests are performed with the valves either open or closed. Due to the large differences observed, "heat traps" were added at a later date.

4.2 Heat Loss Test Method

Heat loss tests for all of the primary and auxiliary water storage tanks are performed to determine the heat loss coefficients. The systems are configured in two different modes for the tests: 1) the flow control valves on the systems are positioned for normal operation (most valves are open, a few are closed) and 2) the valves on the systems are all closed.

4.2.1 Experimental Formulation

The UA values of the tanks are obtained following the procedure given in ASHRAE [95]. The tanks are filled with water at a minimum temperature of 65 °C and are then monitored for at least 12 hours while the tanks are undisturbed. The UA coefficient is

determined by a linear regression on the line obtained by plotting the logarithm of the temperature difference (tank average temperature minus ambient temperature) versus time, which is obtained by integrating eqn (4.1), yielding eqn (4.2).

$$MC_p \frac{d\Delta T}{dt} = UA(\Delta T), \quad \Delta T = \bar{T}_{\text{tank}} - T_{\text{ambient}} \quad (4.1)$$

$$\ln \Delta T = \frac{UA}{MC_p} (t_2 - t_1) + C \quad (4.2)$$

The tank average temperature is calculated as a volume weighted average based on the nodes (temperature measurements) in the tank. An area weighted average is computed for comparison, the area being the outside area of the tank. The nodes are evenly spaced throughout the height of the tank, as shown for an auxiliary tank in Fig. 4.1. The water in the tank becomes stratified over the course of the test, as shown for the drain-back system auxiliary tank in Fig. 4.2. Other temperature profiles for all the tanks and tests are similar. The sharpest temperature drop is at the bottom of the tank. This is not accounted for when calculating an average UA value.

The r^2 (a measure of the total variability in $\ln \Delta T$ that is accounted for when using the line fit to predict $\ln \Delta T$) of the regressions is 0.9 or greater. The regression lines for the main solar and auxiliary storage tanks for the drain-back system for one such experiment is shown in Fig. 4.3. The waviness of the line is due to the cycling on and off of the room heater located in the basement of Solar House I where the tanks are located.

Figure 4.1 Thermocouple Tree in Auxiliary Tank

Figure 4.2 Auxiliary Tank Temperature Profile

Figure 4.3 Tank Heat Loss Coefficient Determination

4.2.1.1 Area Weighting vs. Volume Weighting

The average tank temperature used to calculate the overall UA value for the tank is determined using two methods. The first one weights each node, N , temperature (eight nodes per tank, corresponding to the eight measurements) based on the outside surface area of the tank. This formulation is sensible because the heat loss from the tank is over this area. The second method weights each node temperature by the volume of water con-

tained in that node; which is sensible because it is the temperature of this volume of water used in the temperature difference. The average is obtained as given in eqn (4.3) for each time step. The variable AV_j refers to either area or volume.

$$\left(\sum_{j=1}^N (T_j \cdot AV_j) \right) / \left(\sum_{j=1}^N AV_j \right) \quad (4.3)$$

The results from both methods are almost identical. The average temperature using the volume weighted average is 0.4% to 0.8% higher than the area weighted average temperature. This leads to a volume weighted log temperature difference of 0.2% to 0.4% greater than the area weighted log temperature difference. The calculated overall UA values vary less than 1% between the methods. An area weighted average is used in this dissertation.

4.2.2 Analytical Formulation

The UA value for the water storage tanks is calculated using the R-value of the insulation given by the manufacturer. The film coefficient is based on an average 10 °C temperature difference between the ambient air and the tank surface temperature. The following equations from Welty, Wicks, and Wilson [1976] apply. There are two film coefficients, one to the room ambient, the other to the water inside the tank. These two terms are determined using eqn.'s (4.2) and (4.3) for the sides of the tank, and eqn.'s (4.4) and (4.5) for the top of the tank. The convection to the water contributes a negligible amount.

$$Nu_{water, sides} = 0.548(GrPr)^{1/4} = (hL)/k \quad (4.4)$$

$$h_{air, sides} = 5.678[0.19(\Delta T)^{1/3}], \text{ W/m}^2\text{C} \quad (4.5)$$

$$Nu_{water, top} = 0.27(GrPr)^{1/4} = hL/k, \quad L=0.9D \quad (4.6)$$

$$h_{air, top} = 5.678[0.22(\Delta T)^{1/3}], \text{ W/m}^2\text{C} \quad (4.7)$$

Once the h 's are known, along with the insulation R -value and the conduction from the bottom of the tank through the floor, the overall resistance is calculated as:

$$\frac{1}{R_{tot}} = \frac{A_{sides}}{R_{sides}} + \frac{A_{top}}{R_{top}} + \frac{A_{bottom}}{R_{bottom}} = UA \quad (4.8)$$

$$R_i = \frac{1}{h_{air}} + \frac{1}{h_{water}} + R_{ins}$$

The calculated UA 's are 1 W/°C for all the auxiliary tanks, 2.4 W/°C for the drain-back system solar storage tank, and 1.2 W/°C for the natural convection heat exchanger system solar storage tank. These results are summarized in Table 4.2.

4.2.3 Valves Open/Valves Closed

Tests on the tanks are performed twice, once with the flow control valves in the position for normal operation (most valves in the open position), and a second time with all valves closed. The results from the two tests are compared to determine the degree to which convective currents in the pipes affect the heat loss from the tanks. Both sets of tests were conducted in 1994 (without heat traps) and in 1995 (with heat traps). In addition to

these tests, the natural convection heat exchanger system is tested with the thermosyphon loop closed, and with the collector loop closed.

4.2.4 Heat Traps

Large differences are observed between the UA's when the valves are in the normal operating position and when the valves are closed. In an attempt to reduce the heat loss from the tanks when the valves are in the normal operating position, "heat traps" were added to some pipes going into/out of the hot water tanks, as shown in Figures 4.4, 4.5, and 4.6 (the valves that are in the closed position for normal operation are noted on the figures), the heat traps are effected by a U-turn in the piping with a minimum 0.152 m (6 in.) depth [Wood, 1994; Instructions for Installation and Operation, Solar Water Heaters, 1983]. Heat traps inhibit natural convection of the hot fluid in the vertical riser pipes by forcing the fluid to turn and then directing it downward, against the natural tendency of the hot fluid to rise. Differences between the results from tests with and without the heat traps are attributed to free convective heat loss through the attached pipes. Evidence of convective currents affecting the heat loss is presented by Wood et al., 1995, technical progress report #3. The heat loss from a tank filled with hot water increased by 10% when a 0.457 m (18 in.) pipe, empty of water, was connected to the top of the tank. The heat loss increased by 30% when the pipe connected to the top of the tank was full of water. A summary of test conditions is given in Table 4.1.

Figure 4.4 Drain-Back Instrumentation with Heat Trap Installed

Legend	
T	Temperature Measurement.
P	Power Measurement.
ΔP	Differential Pressure Measurement.
$\dot{V}$	Flow Measurement
	Pump
	Pressure Gauge and Regulator
	Pressure Relief Valve
	Ball Valve
	Check Valve
	Resistive Heater Element
	Solenoid Valve
	Visual Flow Meter

Figure 4.5 ICS Instrumentation with Heat Trap Installed

Figure 4.6 Natural Convection Heat Exchanger Instrumentation with Heat Trap Installed

Table 4.1 UA Test Conditions

Heat Traps:	Valve Position	Start Date & Time	Duration	Tank Initial Temp.
w/o Heat Traps	Valves Open	6/9, 2 p.m.	25 hrs.	65 °C
	Valves Closed	6/7, 5 p.m.	45 hrs.	65 °C
w/ Heat Traps	Valves Open	8/14, 4 p.m.	96 hrs.	70 °C
	Valves Closed	8/2, 1 p.m.	95 hrs.	65 °C

4.3 Results

The heat loss coefficients are given in Table 4.2 for each of the systems tested; also included are the analytically determined heat loss coefficients for each of the tanks. The first set of tests performed were without heat traps installed. The valves were in the open position for the first test without the heat traps, the valves were in the closed position for the second test without heat traps. Heat traps were then added and the valve position tests repeated.

The analytically determined UA value should compare well to the UA determined from the valves closed test, this comparison is shown in Table 4.3. In almost all cases, the analytically determined value is less than the experimental value. This difference varies from 70% to 150%. The one exception is for the primary tank for the drain-back system. The analytical value of 2.4 W/°C is 24% higher than the 'without heat traps' test value of 1.82 W/°C, and only 7% less than the 'with heat traps' test value of 2.56 W/°C.

Table 4.2 UA Results

Watt/deg C	Analy.	W/O Heat Traps		W/ Heat Traps	
		Valves Open	Valves Closed	Valves Open	Valves Closed
Drain-Back: Primary	2.4	2.43	1.82	2.93	2.56
Auxiliary	1	2.40	1.79	2.04	1.69
ICS	1	4.3	2.0	2.6	1.8
Nat. Conv. HX: Primary	1.2	3.67	2.97	5.36	2.82
Auxiliary	1	2.21	1.66	2.06	1.72

Table 4.3 UA Results - Valves Closed Only

Valves Closed Watt/deg C	Analy. (a)	w/o heat traps (b)	w/heat traps (c)	% Diff (a - b)/a	% Diff (a - c)/a	% Diff (b - c)/b
Drain-Back: Primary	2.4	1.82	2.56	24	-7	-41
Auxiliary	1	1.79	1.69	-79	-69	6
ICS	1	2.0	1.8	-100	-80	10
Nat. Conv. HX: Primary	1.2	2.97	2.82	-148	-135	5
Auxiliary	1	1.66	1.72	-66	-72	-4

The analytical value being lower than the measured value can be attributed to one of three factors: (1) the actual losses from the tanks are increased by the pipe extensions, "fins," at the top of the tank (supply and draw lines), (2) since the tanks are stratified (the hottest water being at the top of the tank), the losses from the top, and the "fins," are

greater than would be calculated, (3) and the fact that the average temperature difference and the Nusselt and convective coefficient correlations used for the analytical calculations are not fully representative.

After the heat traps were added, a decrease in the UA of between 5% to 15% occurred for most of the tanks. This is shown in Table 4.3 for the tests with the valves closed (the percentage difference of interest is given in the last column), and Table 4.4 for the tests with the valves open. As expected, the UA decreases more for the tests when the valves are open. These differences are considered significant and are attributed to the heat traps. In comparison, the UA values are decreased 25%, on average, by closing the valves (discussed in a later paragraph).

Table 4.4 UA Results - Valves Open Only

Valves Open Watt/deg C	w/o heat traps (a)	w/heat traps (b)	% Diff (a - b)/a
Drain-Back: Primary	2.43	2.93	-21
Auxiliary	2.40	2.04	15
ICS	4.3	2.6	40
Nat. Conv. HX: Primary	3.67	5.36	-46
Auxiliary	2.21	2.06	7

If the results for the valves open and valves closed tests, without heat traps, are compared, the UA values for all of the tanks decrease; the same is true if the results for the valves open to valves closed tests, with heat traps, are compared. This is shown in Table

4.5. This difference ranges from 15% to 50%, with an average of 25%, and is expected since fluid convection is eliminated by closing the valves. This percentage is greater in all cases but the natural convection heat exchanger primary solar storage tank; since the valves that are closed are located after the heat traps, they have less effect on the convective currents when heat traps are in place than when they are not.

Table 4.5 UA Results - Valves Open/Closed Comparison

Watt/deg C	W/O Heat Traps			W/ Heat Traps		
	Valves Open (a)	Valves Closed (b)	% Diff (a - b)/a	Valves Open (c)	Valves Closed (d)	% Diff (c-d)/c
Drain-Back: Primary	2.43	1.82	25	2.93	2.56	13
Auxiliary	2.40	1.79	25	2.04	1.69	17
ICS	4.3	2.0	53	2.6	1.8	31
Nat. Conv. HX: Prim.	3.67	2.97	19	5.36	2.82	47
Aux.	2.21	1.66	25	2.06	1.72	17

Note the increases in UA from the 'without heat traps' to the 'with heat traps' tests for the primary tanks of both the drain-back and natural convection heat exchanger systems, and the 40% decrease for the ICS tank with the valves open. For the drain-back system, this is probably due to degradation of the insulation wrapped around the tank due to saturation with water (the drain-back tank system is to be upgraded to a new natural convection system which is being tested, so corrective action has not been taken to replace the insulation).

For the natural convection heat exchanger system main tank, the Micro Motion flow meter used in the 'without heat traps' test was responsible for 1/3 of the pressure drop in the system. It was replaced with a Micro Motion flow meter with 1/10 of that pressure drop. An additional heat loss test conducted with the new flow meter removed from the system yielded a UA value of 5.4 W/°C, which is within the experimental resolution of the 'with heat traps' natural convection heat exchanger system test with the valves open. This leads to the assumption that the present Micro Motion flow meter does not restrict the natural convection flow to any considerable degree. Therefore, it is hypothesized that the UA of the natural convection heat exchanger primary tank increased by 46% due to increased recirculation (backflow) through the natural convection heat exchanger loop.

Two other tests on the natural convection heat exchanger system are performed to isolate the effect of recirculation. Two possibilities exist for the extent of the recirculation - through the thermosyphon loop and through the collector loop. In the first test, all valves are in the normal operation mode, except for the valves on the glycol side of the heat exchanger. These valves are closed. The calculated UA is 5.39 W/°C, which, as expected, is for all practical purposes the same. For the second test, all valves are in the normal operation mode except for the valves on the thermosyphon loop, which are closed. The UA determined from this test is 3.88 W/°C, a significant difference of 30%. This leads to the very important conclusion that there is significant backflow through the heat exchanger when the system is idle.

The 40% decrease in UA for the ICS system, with the valves open, is somewhat of a mystery. The plumbing for the ICS system, as seen in Fig. 4.3, is very complex. Initially, it was believed that valves normally in the closed position were left open for the test.

Therefore, after the heat traps were installed, another series of heat loss tests was performed on the ICS system. In one test, all valves remained open, which resulted in a UA value of $2.4 \text{ W/}^{\circ}\text{C}$. In the second test, the valves were placed in their normal operation position (the "valves open" mode). The resulting UA is $2.6 \text{ W/}^{\circ}\text{C}$, an 8% difference. This result is opposite of what is expected, since the second test actually has fewer valves open than the first. The decrease in part can be explained by the new insulation that was installed on the pipes in December of 1994. As stated, the ICS plumbing is very complex, and these results point to the fact that the effects on the UA of water heater tanks of pipes extending from them is not known.

4.4 Conclusions

An accurate determination of the heat loss coefficient for the water tanks in a solar system is important since it accounts for as much as 15% of the net energy gained at the collector. Given this, it is important to perform the test with the necessary plumbing installed since it can increase the UA significantly. The addition of heat traps can reduce the tank losses by 15% of the original. There is considerable backflow on the natural convection heat exchanger system which increases the nighttime losses through the heat exchanger; a very low pressure check valve should be considered for installation in the thermosyphon loop of the natural convection heat exchanger.

CHAPTER 5. HEAT EXCHANGER ANALYSES

5.1 Introduction

Two of the systems being tested use liquid-to-liquid heat exchangers to transfer the heat from the solar generated hot water to mains supply water. The overall performance of the systems depends directly on the performance of these heat exchangers. Analysis of the unpressurized drain-back system, which uses a load side heat exchanger is provided in this chapter. The fluid on both sides of the heat exchanger is water.

The side-arm shell and coil heat exchanger used with the natural convection heat exchanger system has been studied extensively in the literature [Allen, et al., 1994; Avina, et al., 1995; Bergelt, et al., 1993; Davidson, 1994-1996; and Fraser, 1992] and so the analysis is not repeated here.

5.2 Drain-Back System Load Side Heat Exchanger

The performance of the drain-back system load side heat exchanger will be analyzed using two different measures of performance to determine which is the most useful. The first measure is the heat exchanger effectiveness. The second measure is the overall heat transfer resistance of the submerged coil.

Load-side heat exchangers are used in solar domestic hot water heating systems for several reasons. One such reason is to allow the use of an unpressurized storage tank; thus,

a drain-back system (to avoid freezing of the lines to and from the collector) can be used. Also, if the system is unpressurized, the collectors can be designed for lower pressures. Furthermore, the use of a load-side heat exchanger eliminates the need for a pump on the load delivery side. The system tested is shown schematically in Fig. 5.1. To transfer the heat from the solar storage tank to the load, water mains supply, shown as point A, travels downward to point B, and then circulates upward through the coil that is submerged in the solar storage tank. The water exits the solar storage tank and then enters the auxiliary tank, via a dip tube, at point C. This arrangement may promote stratification in the solar storage tank, depending on the location of the coil, which can increase collector performance [Farington and Bingham, 1987]. This type of heat exchanger operates only when a draw is occurring and is a "one-pass" system. In support of the Solar Ratings and Certification Corporation's (SRCC) ongoing work, validation of the model of the load-side heat exchanger is important.

5.2.1 Heat Exchanger Description

The test data presented here were acquired during the time period of July 24 - 28, 1994. The test procedure is normal operation. The results from other testing periods are similar. This system is described in Chapter 2, section 2.2.3.

The load-side heat exchanger coil is 36.58 m (120 ft.) of 0.0191 m (3/4 in.) copper tubing coiled through nearly the entire height of the solar storage tank. The coil is situated near the outer edge of the inside of the tank. The draw flow rate is $1.96 \times 10^{-4} \text{ m}^3/\text{s}$ (3.1 gpm), a turbine flow meter with an accuracy of $\pm 0.5\%$ is used to measure the flow.

Figure 5.1 Weather Station and System Instrumentation

5.2.2 Experimental Results

Two days of the five days of test data will be presented. The two days are July 26 and July 27, 1995. These days are representative of the entire data set. July 26 was a fairly cloudy day and July 27 was a clear day with the highest solar insolation of the five days. As an indirect example of the heat exchanger performance, the solar storage tank temperatures are shown versus time in Fig. 5.2. The top node of the tank is denoted by $N=1$, while the bottom node of the tank is denoted by $N=8$. The times when draws are performed are denoted on the figures with a "D."

An average stratification of 10-15 °C is observed in the tank, with greater stratification in the afternoon, due to the solar contribution. The very top node of the tank is cooler than several of the nodes below it. This is due mostly to an air gap which exists at the top of the tank when the collector is being pumped, and in part to the heat loss out the top of the tank. The stratification becomes more pronounced during a draw because the temperature in the bottom of the tank decreases by as much as 15 °C while the temperature in the top of the tank decreases by only 3-5 °C. This suggests that the mains supply water gains most of its energy from the bottom of the tank. During the day, between draws, the tank water temperature increases due to the solar contribution. This is especially noticeable in the bottom of the tank (probably due to the stratification manifold). During the night, the bottom of the tank actually warms due to the conductive heat transfer from the rest of the tank.

Fig. 5.3 shows the time histories of the temperature rise across the heat exchanger. The water mains supply temperature remains fairly constant at 16 °C. As seen from Fig. 5.3, initially the energy gain from the heat exchanger is very high. This is due to the fact

that the fluid in the heat exchanger has had the time in between draws (at least four hours) to come into equilibrium with the temperature of the fluid in the tank. During a draw, the residence time of water in the coil is 53 seconds at a draw flow rate of $1.96 \times 10^{-4} \text{ m}^3/\text{s}$ (3.1 gpm). Thus, after about 1 minute, the heat exchange should begin to be well represented by traditional heat transfer/exchanger correlations. The additional amount of energy in the “humps” in Fig. 5.3, totaled over the day, is only three percent of the total daily energy drawn off.

Figure 5.2 Main Solar Storage Tank Temperatures

5.2.3 Analysis

The data are analyzed to determine two parameters, the effectiveness and the overall heat transfer coefficient. The errors reported in the text are computed in quadrature as

Figure 5.3 Heat Exchanger Inlet and Outlet vs. Time

recommended by Holman [1978]. The fluid properties are assumed to be constant and known.

5.2.3.1 Effectiveness

In this section, the performance of the heat exchanger is considered based on heat exchanger design theory. The average heat exchanger effectiveness is calculated from the experimental data and a representative correlation that best predicts the experimental effectiveness is sought.

The heat exchanger effectiveness is calculated at each data point using eqn (5.1).

$$\epsilon = \frac{\Delta Q}{\Delta Q_{\max}} = \frac{\dot{m}_{\text{drw}} C_p (T_{\text{out, HX}} - T_{\text{in, HX}})}{\dot{m}_{\text{drw}} C_p (T_{\text{TnkMax}} - T_{\text{in, HX}})} \quad (5.1)$$

Figure 5.4 presents the calculated effectiveness for over two days of draw data. The average effectiveness is obtained by summing all of the individually calculated effectivenesses and dividing by the total number of data points. The average effectiveness for all data is 0.78 +/- 0.5%. This excludes the first minute of data when the heat exchanger is being purged of the warmed coil water. The disparity in the curves is due to the dependence of the outside convective heat transfer coefficient on the outside temperature difference. Notice on Fig. 5.2 that the tank temperature does not rise significantly between the 8 a.m. draw and the noon draw on July 26, 1995, and that these temperature data points are the lowest on the plot. This was a cloudy day and the pump did not operate much; hence the temperature difference is lower, resulting in a reduced Raleigh number, and thus a lower effectiveness. Also, it was found that the effectiveness is not significantly affected by whether or not the pump is on or off during the actual draw. This is not surprising since the tank is so large.

The effectiveness is predicted using various effectiveness correlations for different types of heat exchangers in order best to match the experimental effectiveness. The data indicate that the operation of this heat exchanger most resembles a parallel, counterflow shell and tube, shell fluid mixed; or a cross flow, one fluid mixed type heat exchanger. The minimum capacity rate is based on the draw flow (817 W/°C) and the maximum capacity rate is based on the tank capacitance (3428 W/°C). Since $C_{\max}$ is much greater than $C_{\min}$, the flow arrangement configuration becomes less important in the effectiveness calculation. Equation (5.2) [Kays and London, 1964] is for a cross flow heat exchanger with one

Figure 5.4 Heat Exchanger Effectiveness

fluid mixed. The average overall heat transfer coefficient, UA , used is $1500 \text{ W/}^\circ\text{C}$, calculated from experimental data (discussed later). The resulting calculated effectiveness is 0.76.

$$\epsilon = \frac{C_{\max}}{C_{\min}} \left(1 - e^{-\Gamma \frac{C_{\max}}{C_{\min}}} \right), \quad \Gamma = \left(1 - e^{-N_{TU}} \right), \quad N_{TU} = \frac{UA}{C_{\min}} \quad (5.2)$$

5.2.3.2 Heat Transfer Coefficient Models

In this section, the heat exchanger performance is considered based on the overall heat transfer coefficient. Four different models are used to predict the outside heat transfer coefficient of the heat exchanger (the heat transfer coefficient between the outer wall of the

heat exchanger coil and the water in the hot water tank). None of these models strictly apply to this heat exchanger because of the different assumptions and configurations of the models. These are then used to calculate an overall heat transfer coefficient. The predicted coefficients are then compared to the experimentally determined average values.

The overall heat transfer resistance of the heat exchanger is comprised of three terms. The first is the inside convective coefficient due to forced flow through the coil (draw flow), the second is the conductance through the copper tubing, and the last is the outside convective heat transfer coefficient due to the natural convection flow around the coil.

The inside convective coefficient, h_i , is determined using the Dittus and Boelter equation, eqn (5.3), [Welty,1976], for the fluid being heated. The Reynolds number is 19,104 (the flow is turbulent), yielding a Nusselt number of 109.

$$Nu_i = \frac{h_i D}{k} = 0.023 Re_D^{0.8} Pr^{0.4} \quad (5.3)$$

The corresponding convective coefficient is 3,576 W/m² °C (630 Btu/hr ft² °F) +/- 7%. This correlation is well documented in the literature, but it is for a horizontal straight coil. The curvature will increase the convective coefficient by a factor of 1.16 +/- 10%, Chauvet et al., [1994]. Thus, h_i is taken as known because it is well documented, and it is much larger than h_o and therefore not as important in determining the overall heat transfer. The resistance through the copper tubing is negligible. Therefore, only the outside convective coefficient remains to be calculated. Once the outside convective coefficient is determined, based on the outside surface area ($\pi D_{\text{tube}} L_{\text{coil}}$) of the coil, A_{coil} , the overall heat

transfer resistance is calculated from eqn (5.4). The values of A_o and A_i are given below for each correlation used to calculate the convective coefficient.

$$U_{tot} = \left[A_{coil} \left(\frac{1}{h_o A_o} + \frac{1}{h_i A_i} \right) \right]^{-1} \quad (5.4)$$

The outside convective coefficient is predicted using four methods. Method (a) uses a correlation for a vertical cylinder, given as eqn (5.5) from Kakac [1987], which is valid for all Ra_D . A Nusselt number is calculated for both the inside and the outside of the vertical cylinder. For the outside, the large coil diameter (0.66 m, 26 in.) is used as the characteristic length, D , and is also used to determine A_o . For the inside, the smaller inside coil diameter (0.622 m, 24.5 in.) is used for D and in determining A_i . This correlation actually is for an isothermal solid wall in an infinite fluid, with no influence from the surrounding wall; none of these restrictions strictly apply to this case.

$$Nu = \frac{4}{3} \left[\frac{7GrPr^2}{5(20 + 21Pr)} \right]^{\frac{1}{4}} + \frac{4(272 + 315Pr)L}{35(64 + 63Pr)D} \quad (5.5)$$

Method (b) assumes that the coil is a vertical flat plate (front and back). This could be inferred since $D/L > 35/Gr_D^{1/4}$ [Kakac, 1987], with less than 5% error. The correlation, given as eqn (5.6), is based on the height of the tank, L (1.4 m, 55 in.). The same Nusselt number is used for both "sides" of the flat plate with A_o and A_i as for eqn (5.5). These first two methods do not account for the interaction of the fluid leaving one coil loop and interfering with the next coil loop, nor for the effect of flow over a cylinder.

$$Nu = \left\{ 0.825 + \frac{0.387Ra^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2 \quad (5.6)$$

Method (c) is for flow over a horizontal cylinder; this method also does not account for the interactions between coil loops. The Rayleigh number is based on the diameter of the tubing (0.019 m, 0.75 in.) and A_o and A_i are both equal to A_{coil} . The McAdams correlation [Welty et al., 1976], which holds for $10^4 < Ra_D < 10^9$, is used, eqn (5.7). The exponent of 1/4 is for laminar flow.

$$Nu = 0.53(Gr_D Pr)^{1/4} \quad (5.7)$$

Although all of these equations may be plotted versus temperature difference to yield a relationship, it is desired to determine which correlation lies in the neighborhood of the data. Therefore, in all three of the above methods, an average representative temperature difference is used. This temperature difference is calculated to be 14.7 °C (the minimum was 10.8 °C and the maximum was 22.5 °C. The temperatures during the purging time are not included). This is an average of the difference between the average tank temperature and the coil temperature ($T_{out,HX}/2 + T_{in,HX}/2$), calculated at each instant in time over the course of a draw. This average temperature difference is termed "coil dT."

Method (d) uses the correlation from Farrington and Bingham [1987], eqn (5.8). Note that they recommend using a Log Mean Temperature Difference (LMTD), given as eqn (5.9), as the driving potential. The calculated average LMTD is 9.8 °C (the minimum was 6.5 °C and the maximum was 14.1 °C; the LMTD's during the purging time are not included). Further, they recommend $C=0.9$ and $m=1/4$ (for laminar flow).

$$Nu = CRa_D^m \quad (5.8)$$

$$LMTD = \frac{(T_{TnkBot} - T_{in, HX}) - (T_{TnkMax} - T_{out, HX})}{\ln \left(\frac{T_{TnkBot} - T_{in, HX}}{T_{TnkMax} - T_{out, HX}} \right)} \quad (5.9)$$

The results using the above correlations are presented in Table 5.1, along with the results obtained experimentally, which are presented in the next section.

5.2.3.3 Experimental Data Reduction

The test results are reduced using three different methods. Equation (5.10), method (e), uses the measured values of the temperature difference across the coil to do an energy balance on the coil, for M data points. Since three draws are performed on each of the 5 days, for a total of seven minutes per draw, and the data are taken every 8 seconds, the average is obtained for about M=262 data points. Method (f) uses the same equation, except the temperature difference, (coil dT), is replaced with the LMTD. In all cases, the area, A, is the total submerged surface area of the coil and is equal to 1.86 m² (20 ft²). The average U value calculated is 1,500 W +/- 2.5%, using the coil dT. Once the overall heat transfer coefficient is known, the average external convective heat transfer coefficient can be determined to within +/- 5%.

$$U_{meas} = \left(\sum_{i=1}^M \frac{\dot{Q}_{drw,i}}{A(coil \, dT)_i} \right) / M = \left(\sum_{i=1}^M \frac{(\dot{m}c_p)_{drw,i} (T_{out,HX} - T_{in,HX})_i}{A(coil \, dT)_i} \right) / M \quad (5.10)$$

Equation (5.11), method (g), satisfies the heat balance equation on each node, j , of the entire heat exchanger. The heat exchanger coil is treated as a 36.58 m (120 ft.) long tube. The distance traveled along the tube in the direction of flow is designated as x , and P is the perimeter of the tube. At each node j of the tank the temperature of the fluid inside the tube is $T_{x,j}$. The temperature of the surrounding tank water at each node j is $T_{Tank,j}$. Here, the tank and tube are divided into eight equal segments ($j=1$ to N) to correspond to the eight tank temperature measurements. A constant overall heat transfer conductance, U_{tot} , is then assumed for all nodes. For each time step, i :

$$\ln \Delta T_{out,j} = \ln \Delta T_{in,j} - \frac{U_{tot,i} P}{\dot{m} C_p} x, \quad \Delta T_j = T_{x,j} - T_{Tank,j} \quad (5.11)$$

$$U_{tot} = \left(\sum_{i=1}^M U_{tot,i} \right) / M$$

For example, applying eqn (5.11) for $j=1$, $\Delta T_{in,1}$ is equal to the measured heat exchanger inlet temperature minus the bottom tank node temperature, yields the inlet ΔT to node 2, as shown in eqn (5.12). This is repeated successively until the outlet temperature of the coil at $j=N$ is obtained. The process is repeated using different U_{tot} values until the final calculated coil outlet temperature, $T_{out,N}$, predicted matches the experimentally measured heat exchanger outlet temperature. This process is repeated for each data point,

M, resulting in M predictions for U_{tot} ; the average of these over time is then taken for the final value.

$$\begin{aligned}
 \ln \Delta T_{out,1} &= \ln \Delta T_{in,1} - \frac{U_{tot,i} P}{\dot{m} C_p} x, & \Delta T_{in,1} &= T_{main, HX} - T_{TnkBot} \\
 \ln \Delta T_{out,2} &= \ln \Delta T_{out,1} - \frac{U_{tot,i} P}{\dot{m} C_p} x \\
 &\vdots \\
 \ln \Delta T_{out,N} &= \ln \Delta T_{out,N-1} - \frac{U_{tot,i} P}{\dot{m} C_p} x, & \Delta T_{out,N} &= T_{measout, HX} - T_{TnkTop}
 \end{aligned} \tag{5.12}$$

The results of all the calculations are provided in Table 5.1. The % diff. column represents the percentage difference in the total heat transfer as computed to that calculated experimentally. The experimental average heat transfer value, eqn. (5.13) is 22,099 W +/- 1%. Note that the horizontal cylinder correlation, method (c), agrees the best with the measured data to within 23%. Since the correlation for a single horizontal cylinder underpredicts the heat transfer, stacking the coils seems to enhance the heat transfer. As the coils within the tank are stacked on top of one another with almost no intervening gap, the enhanced heat transfer may be attributed to vortex shedding off of each coil, with this local turbulence increasing the heat transfer.

$$\dot{Q}_{expave} = \left\langle \sum_{i=1}^M \dot{m} C_p (T_{out, HX} - T_{in, HX})_i \right\rangle / M \tag{5.13}$$

Table 5.1 Heat Transfer Properties

Method	Eqn	Ra	h_o (W/ m ² C)	UA (W/C)	Q (W)	% diff.
(a) vert. cyl.	(2)	1.5×10^{11}	315	414	6,105	72
(b) vert. plate	(3)	1.4×10^{12}	672	825	12,162	45
(c) horiz. cyl.	(4)	3.6×10^6	756	1,159	17,082	23
(d) Farr&Bing	(5)	3.4×10^5	716	1,113	10,909	51
(e) $Q_{ave}/A(\text{coil } dT)$	(7)	3.4×10^6	1,042	1,504	22,173	-0.3
(f) $Q_{ave}/A(\text{LMTD})$	-	3.4×10^5	1,852	2,264	22,188	-0.4
(g) exponential	(8)	3.4×10^6	1,031	1,493	22,004	0.4

Also, note the poor agreement (51%) when using Farrington and Bingham's correlation, method (d). This is perhaps due to the short duration of our draws (7 minutes) relative to the long duration of Farrington and Bingham's tests (3 to 4 hours). Only a small fraction of energy is extracted from the tank over 7 minutes, a "shallow" draw, while in contrast in Farrington and Bingham's tests the tank was nearly completely purged of energy over 4 hours, a "deep" draw. Thus, the performance of our system, where the tank temperature changes little, is not modeled well using the Farrington and Bingham correlation. Since our tank temperature changes little, and the heat exchanger outlet coil temperature is still many degrees cooler than the tank temperature, it is concluded that a load-side heat exchanger of this design appears to be relatively ineffective at extracting energy from the tank over "shallow" draw profiles. Therefore, it probably leads to sub-optimal performance of the system.

5.2.3.4 Curve Fits

The generalized form of eqn (5.8) is now re-evaluated using the experimental data to determine the Nusselt - Raleigh correlations. The correlation is considered for both average representative temperature differences: the coil dT (14.7 °C) and the LMTD (9.8 °C). The logarithm of the average Nusselt number per draw is shown in Fig. 5.5 plotted against the logarithm of the average Raleigh number per draw. Each line represents different values for C and m, as given in eqn (5.14), obtained by a least squares fit to the data shown in Fig. 5.5. For one case, the coefficient m is forced to be 1/4, for laminar flow, and the average C value is then determined. For the second case, a two-parameter curve fit is performed, thus both m and C determined. The correlation of Farrington and Bingham is also presented.

$$\text{Log}_{10}(\text{Nu}) = \text{Log}_{10}(C) + m\text{Log}_{10}(\text{Ra}) \quad (5.14)$$

As can be seen in Fig. 5.5, using the LMTD as the representative temperature difference results in much greater spread of the data. Hence, the coefficient of determination, r^2 , for these fits has a lower value than when using the coil temperature difference. The calculated slopes, m, for both models are lower than for laminar flow (0.104 using the coil temperature and 0.188 using the LMTD). However, given the limited range of the Raleigh numbers, the validity of the slopes is questionable.

The calculated heat transfer rates using these results are shown in Table 5.2. By using the coil temperature difference, the final result is within 2% of the measured value, regardless of whether m is forced to be 1/4 or not. If the LMTD is used, the difference

increases to 8%. Therefore, it is recommended that the coil temperature difference be used as the driving force for load-side heat exchangers if “shallow” draws are taken.

Table 5.2 Heat Exchanger Coefficients

TEMP. DIFF.	m	C Value (ave)	Ra	h_o (W/m ² C)	UA (W/C)	Q (W)	% diff.
coil dT m forced	0.25	0.75	3.4×10^6	1,067	1,533	22,598	-2
coil dT curve fit	0.104	6.66	3.4×10^6	1,048	1,513	22,300	-1
LMTD m forced	0.25	2.60	3.4×10^5	2,064	2,442	23,934	-8
LMTD curve fit	0.188	5.65	3.4×10^5	2,041	2,425	23,764	-7
LMTD Farr&Bing	0.25	0.9	3.4×10^5	716	1,113	10,909	+51

Figure 5.5 Log(Ra) vs. Log (Nu)

5.3 TRNSYS Simulation Results

It is also important to assess the performance of the solar system vis a vis the SRCC efforts, which seek to rate system performance using TRNSYS [Klein et al., 1994] simulations [Burch et al., 1993]. One of the overall goals is to match the model to experimental data by adjusting the model parameters to represent more accurately the physical properties of the system.

A TRNSYS sub-program for modeling a load-side heat exchanger does not exist in version 13.1 (used at the time of this work), so the tank models were “tricked” into simulating one by including a heat exchanger effectiveness parameter. The TRNSYS tank model variables are shown in Fig. 5.6 together with a schematic of the actual system. Equation (5.15) is an energy balance for the main and auxiliary tanks of the actual system, and eqn (5.16) is an energy balance for the main and auxiliary tanks of the TRNSYS model system and includes the effectiveness.

$$\begin{aligned} \dot{m}_{pmp} C_p (T_{in, MTnk} - T_{out, pmp}) - \dot{m}_{drw} C_p (T_{in, aux} - T_{sup}) - UA \Delta T_M &= MC_p \frac{dT}{dt}_M \\ \dot{m}_{drw} C_p (T_{drw} - T_{in, aux}) - UA \Delta T_A - Q_{aux} &= MC_p \frac{dT}{dt}_A \end{aligned} \quad (5.15)$$

$$\begin{aligned} \dot{m}_{HM} C_p (T_{HM} - T_{NM}) - \dot{m}_{LM} C_p (T_{1M} - T_{LM}) - UA \Delta T_M &= MC_p \frac{dT}{dt}_M \\ \dot{m}_{HA} C_p (T_{HA} - T_{NA}) - \dot{m}_{LA} C_p (T_{1A} - T_{LA}) - UA \Delta T_A - Q_{aux} &= MC_p \frac{dT}{dt}_A \end{aligned} \quad (5.16)$$

$$\epsilon = \frac{T_{in, aux} - T_{sup}}{T_{Mtnktp} - T_{sup}}, \quad T_{in, aux} = \epsilon(T_{1M} - T_{sup}) + T_{sup}$$

By comparing the above two equations, the temperature and mass flow rate variables in the TRNSYS model can be equated to the appropriate actual system variables, as shown in the figure. The measured average heat exchanger effectiveness of 0.78 is used to adjust the load flow out of the solar storage tank, $\dot{m}_{LM}$. The temperature of the replacement fluid into the solar storage tank, T_{LM} , is at mains supply temperature since the flow into the main solar storage tank from the "hot source" (collectors), T_{HM} , is not equal to the solar storage tank load flow. The flow from the "hot source" (the solar storage tank) into the auxiliary storage tank, $\dot{m}_{HA}$, is zero. The temperature entering the auxiliary storage tank as replacement fluid, T_{LA} , is also adjusted by the effectiveness. The draw flow from the auxiliary tank, $\dot{m}_{LA}$, is the experimental value.

The results for energy are shown in Table 5.3, indicating excellent agreement. Here, the $F_R(\tau\alpha)$ and $F_R U_L$ (first order equation is used) used in the simulation have been determined from the experimental data; more detail is given in Chapter 6. An energy balance was performed on the system using the measured data and closure was obtained to within 2%. An energy balance using TRNSYS predicted values exhibits closure to within 2.6%. A TRNSYS simulation was used to predict the performance for another set of experimental data and the results were equally good. Therefore, this is deemed an acceptable model for this system.

Table 5.3 Measured and TRNSYS Results

Quantity	Measured _i	TRNSYS	%Error
Q_{useful} (MJ)	187	186	+0.3
Q_{draw} (MJ)	114	103	+9.6

(a) Actual System

(b) TRNSYS Model

Figure 5.6 TRNSYS Tank Model

5.4 Conclusions

Analytical calculations for the effectiveness and convective heat transfer coefficients from correlations have been compared against the experimental data. Three models for the convective heat transfer coefficients were analyzed and yielded results that under-

predict the measured heat transfer from 23% to 72%. TRNSYS simulations were performed using the average effectiveness (the calculated effectiveness varies from 0.68 to 0.95); the results compare favorably with experimental results, indicating that a constant effectiveness is an adequate model for the system.

The external flow around the coil is considered to be mostly laminar, natural convection, with local vortex shedding at the coil, while the inside is turbulent, forced convection. The inside heat transfer coefficient contributes little to the overall thermal resistance, while the outside resistance is dominant. The best agreement for the outside heat transfer coefficient is obtained using the correlation for free convection from a horizontal cylinder, with a 23% enhancement most probably due to an increase in the heat transport attributed to turbulence generated from flow over the stacked coils of the heat exchanger. An average effectiveness of 78% was calculated from experimental data. When used in a TRNSYS model, agreement to within 0.5% of collector useful energy is achieved, as well as agreement to within 10% for the energy delivered, indicating that for "shallow" energy draws, a constant effectiveness model yields good agreement with observations.

"Deep" draws effectively extract more energy from the main solar storage tank as opposed to the "shallow" draws of this system. Therefore, a design improvement is to use a natural convection heat exchanger that would be able to continuously extract energy from the main solar storage tank and deliver it to the auxiliary storage tank for use during a draw.

CHAPTER 6. SOLAR COLLECTOR PARAMETER ESTIMATION

6.1 Introduction

The collector of a solar domestic hot water system is the most important component. Its performance directly affects how much useful energy will result from the incident solar radiation. The design process should result in the economically optimum efficiency based on the life cycle cost of the systems. In order to optimize analytically, though, models for the operation of the collectors must be available.

Also, since every collector can not be tested for every foreseeable operating condition, models are needed which can predict the performance (hence, give a rating) of collector-based data measurements which can be easily obtained. These ratings are important to both the manufacturer and the consumer. So, if the ratings procedure can be verified and/or improved, greater confidence in the ratings of solar domestic hot water systems may increase the industry's market.

The analyses of the two flat-plate collectors tested is presented in this chapter. The two flat-plate collectors tested are analyzed to determine the parameters most important to the collector's performance and to assess the current models used for ratings of flat-plate collectors. This analysis begins with the most fundamental performance equation, a first-law energy balance on the system; from this the Hottel-Whillier-Bliss (HWB) equation is deduced. The analysis then delves into advanced performance equations using additional

parameters. The performance under typical operating conditions is assessed and more accurate performance models are developed.

6.1.1 General Collector Equation

A simplified energy balance, including storage, performed on a flat-plate collector results in eqn (6.1). The rate of energy gain by the system is equal to the absorbed radiation energy, minus the energy collected by the fluid flow through the collector and the convective, conductive, and radiative losses to the ambient. $(MC_p)_e$ is the effective capacitance of the collector, S is the absorbed solar flux, T_p is the mean plate temperature, and U_L is the overall heat loss coefficient of the collector.

$$(MC_p)_e \frac{dT}{dt} + \dot{m}C_p(T_{out,col} - T_{in,col}) = SA_c - A_c U_L (\bar{T}_p - T_a) \quad (6.1)$$

Reference will be made throughout this chapter to the most basic collector model, eqn (6.2). Equation (6.2) compares directly to eqn (6.1) except eqn (6.2) does not include a capacitance term, and the average plate temperature, T_p , has been replaced with $T_{in,col}$. To account for the additional losses incurred since T_p is hotter than $T_{in,col}$, the collector heat removal factor, F_R , is used. It may be considered equivalent to an effectiveness for a heat exchanger, which is the ratio of the actual useful energy to the maximum useful energy that may be obtained. The maximum effectiveness occurs when the entire collector is at the inlet fluid temperature, thus keeping the heat losses to the ambient to a minimum. Note that, through the use of F_R , losses may still exist when the collector inlet temperature is equal to the ambient temperature.

$$\dot{Q}_{useful} = \dot{m}C_p(T_{out, col} - T_{in, col}) = A_c F_R [S - U_L(T_{in, col} - T_a)] \quad (6.2)$$

All of the variables in eqn (6.2) except for $F_R S$ and $F_R U_L$ may be easily measured. Here, $F_R S$ and $F_R U_L$ may be determined from a least squares fit of the data. Dividing by the total incident energy, $A_c G_T$ (collector area times incident radiation on the tilt), results in the collector efficiency:

$$\eta_{col} = \frac{\dot{Q}_{useful}}{A_c G_T} = F_R \frac{S}{G_T} - F_R U_L \left(\frac{T_{in, col} - T_a}{G_T} \right) \quad (6.3)$$

The models developed in this chapter begin with this basic equation. Improvements are made to it by extending S and $F_R U_L$ as described in the following sections.

6.1.2 Collector Capacitance

Note that in eqn (6.2) the heat storage term (from eqn (6.1)) has been dropped. Siebers [1975] indicates that the capacitance of the collector has very little effect on the collector efficiency and useful energy gain. Klein [1973] suggests that including capacitance in the collector equation has two effects. The first is called the "storage" effect and is a pre-heating of the fluid in the collector by the low early morning incident radiation. The additional energy to "storage" was seen to add between 1% to 5% to the useful energy gain. The second effect is termed the transient effect and is defined as the difference in energy gain between a zero capacitance collector and a finite capacitance collector. Klein's results show that a transient effect results from fluctuations in the wind speed, but not from fluctuations in radiation intensity, and that the overall effect on collector performance is

negligible. The work by Wijesundera [1978] shows that a zero capacitance collector model predicts collector useful energy to be 2% higher than a capacitance model using 10 minute averaged weather data, but only 0.5% higher than a capacitance model using hourly averaged weather data. The zero capacitance model will overpredict the energy when the radiation values have a step change from low to high values, but underpredicts the energy when the radiation steps from low to high. Therefore, the overall energy prediction may be acceptable, but the predicted outlet temperature history may be incorrect.

In order to increase confidence of neglecting the collector capacitance, the time constant of the collectors is calculated. The time constant is defined as the time required for the collector outlet temperature to change to $1/e$ of its final, steady value after a step change in incident radiation or inlet fluid temperature. The time constant is found by taking the LaPlace transform of eqn (6.1), yielding eqn (6.4). In this case, the mean plate temperature is taken as the average between the collector inlet and outlet temperatures, assuming a linear spatial profile.

$$(MC_p)_e s \delta T_{out, col} = SA_c - \dot{m} C_p (\delta T_{out, col} - \delta T_{in, col}) - A_c U_L (\delta T_{out, col} + \delta T_{in, col} - 2\delta T_a) \quad (6.4)$$

By grouping like temperature terms, the time constant is determined to be:

$$\tau_c = \frac{(MC_p)_e}{\frac{A_c U_L}{2} + \dot{m} C_p} \quad (6.5)$$

The effective capacitances of the two flat-plate collectors are determined using the weighting factors recommended by Klein [1973]. A weighting factor of 0.2 is used for the

cover. The plate, tube, and headers all have weighting factors of 1.0. And a weighting factor of 0.1 is used for the insulation. All of the terms in eqn (6.5) may be computed approximately by knowing the collector materials and geometry. The properties used for the different materials are obtained from the *ASHRAE Handbook of Fundamentals* [1989]. The collector flow rate is that from experimental values.

The time constant for the natural convection heat exchanger system is less than 3 minutes, and the time constant for the drain-back system is less than 2 minutes. Both of these time constants are less than the averaged data time steps used for the computations. According to Klein et al. [1974], using meteorological data at time steps greater than the time constant of the collector does not permit the full transient effects of performance to be evaluated. However, these effects have been shown to be unimportant. In contrast, the time constant for the hot water tanks, $(MC_p/UA)_{\text{tank}}$, is on the order of 35 hours. Therefore, a steady-state approach for the collectors may justifiably be used in this dissertation.

6.2 Problem Identification

A flat-plate collector consists of tubes attached to an absorber plate enclosed in an insulated box with a clear cover. Fluid flows through the tubes, increasing in temperature through contact with the absorber plate. The cover reduces convective and longwave radiative losses.

Presently, the performance of collectors (included in the SRCC ratings program) is determined from limited experimental data obtained from testing the flat-plate collectors according to ASHRAE Standard 19-1987, which defines very specific procedures. The tests for thermal efficiency must be conducted at near-normal incidence angle, under

steady state conditions, and within specific ranges of ambient temperature, wind speed, collector inlet temperatures, collector flow rates, and incident radiation levels. The performance equation is defined similar to eqn (6.2) except S is replaced by $G_T(\tau\alpha)_{ave}$ (the incident radiation on the tilt times a term to account for the optical characteristics of the collector). A solar simulator may be used for the tests. The collected data are then used to determine the two performance parameters, $F_R(\tau\alpha)_{ave}$ and $F_R U_L$.

In addition, an incidence angle modifier, $IAM = K_{\tau\alpha} = (\tau\alpha)/(\tau\alpha)_n$, must be determined experimentally for off-normal incidence; the collector is mounted such that the incident angles (θ) between the collector and the direct solar radiation are at 0, 30, 45, and 60 degrees, and data are collected with the inlet temperature equal to the ambient temperature so that there is no loss term. $(\tau\alpha)_{ave}$ is then replaced by $IAM_{ave}(\tau\alpha)_n$.

Since it would be impossible to test every collector under every possible operating condition, a few collectors of a given brand and type are tested and the performance parameters are determined. These parameters are then used for every such system when simulations (generally TRNSYS simulations) are performed.

The ASHRAE testing protocol specifies rigid and limited test conditions which may not be realistic to the typical operation of a collector. The parameters determined are taken as constant for all weather conditions even though they are determined for very high incident radiation levels. In this chapter the ASHRAE specified collector performance parameters are determined in-situ, under normal operating and "real" weather conditions. The collector model is then refined to assess the degree to which including more performance parameters yields a better model. The results are also compared to the published SRCC [December, 1994] ratings for these same two systems.

6.3 Theoretical Formulation

The solar radiation absorbed by the collector plate is directly related to the optical properties of the cover and the absorbing capacity of the plate. This quantity is on the order of 70% to 80% of the incident radiation. The collector loss is the compliment of the product of the fraction of absorbed energy and the efficiency of the collector. For many collectors, the efficiency is on the order of 50%; hence, the collector losses to the environment are approximately 20% to 30% of the incident radiation. Therefore, the collector optics are the most important property, and the heat loss properties are secondary. The work by Arafa [1978] corroborates this statement. He found that the collector performance is dependent on the ambient conditions, which are variable, but that the largest influence on the efficiency is the properties of the cover and plate with the dependence of efficiency on wind speed being almost minimal.

Therefore, consideration will be given first to the optics term. As stated in section 6.2, the absorbed solar radiation, S , is equivalent to $G_T(\tau\alpha)_{ave}$. G_T is the incident radiation on the tilt and $(\tau\alpha)_{ave}$ encompasses the optics of the collector. Both of these terms, G_T and $(\tau\alpha)_{ave}$, may be extended to include additional components.

G_T is comprised of three components of radiation incident on the tilt, beam (G_{bT}), sky diffuse (G_{skyT}), and ground reflected (G_{gndT}). G_T may be determined by measuring the actual incident radiation on the tilt, or it may be calculated from the total horizontal radiation, G_H , using radiation models. If pyranometers are not available for data measurement, G_H is more readily available than G_T for various locations, from data obtained through the National Weather Service, such as Typical Meteorological Year (TMY) data.

$(\tau\alpha)_{ave}$ is a transmittance-absorptance product for the collector. τ is the transmittance of the cover, and α is the absorptance of the collector plate. $(\tau\alpha)_{ave}$, as G_T , may be considered to be comprised of three components $(\tau\alpha)_b$, $(\tau\alpha)_{sky}$, and $(\tau\alpha)_{gnd}$, corresponding to each of the radiation components.

Three components for S result from appropriately combining the different radiation components with the different $(\tau\alpha)$ components. These are provided in section 6.4.1. First, the determination of the components of incident radiation is discussed in section 6.3.1 and a discussion of $(\tau\alpha)$ is presented in section 6.3.2.

The second term to be considered is the collector heat loss. The heat loss coefficient, $F_R U_L$, is a complicated function of the overall heat loss coefficient as well as other factors. $F_R U_L$ is discussed in section 6.3.3.

6.3.1 Radiation Processor

The separation of G_H into different tilt components by radiation models is performed in two steps. First, G_H is separated into a beam and sky diffuse component on the horizontal, G_{bH} and G_{skyH} , respectively, through the use of a horizontal radiation model. Second, G_{bH} and G_{skyH} are then projected onto the tilt surface, resulting in G_{bT} , G_{skyT} , and G_{gndT} through the use of a tilt radiation model. G_T is summarized in eqn (6.6) where R_b is the ratio of the beam radiation on the tilt to the beam radiation on the horizontal, R_{sky} is the ratio of the sky diffuse radiation on the tilt to the sky diffuse radiation on the horizontal, and R_{gnd} is the ratio of the ground reflected radiation on the tilt to the total radiation on the horizontal.

$$G_T = G_{bH}R_b + G_{skyH}R_{sky} + G_H R_{gnd} \quad (6.6)$$

The Type 16 radiation processor in TRNSYS has available for use five horizontal radiation models and four tilt radiation models, listed in Tables 6.1 and 6.2. The input to the Type 16 radiation processor is the total horizontal radiation measurement. If just a horizontal radiation model is used, the output is G_{bH} and G_{skyH} . If both horizontal and tilt radiation models are used, the output is G_{bT} and G_{diff} (G_{diff} is equal to G_{skyT} plus G_{gndT}). The radiation models in TRNSYS are standard radiation correlations which may be found in Duffie and Beckman [1991].

Table 6.1 TRNSYS Horizontal Radiation Models

Horizontal
Liu & Jordan (k_T)
Boes (k_T , Sc)
Erbs (k_T)
Reindl - Reduced (k_T , α_{sa})
Reindl - Full (k_T , α_{sa} , T_a , ϕ)

Table 6.2 TRNSYS Tilt Radiation Models

Tilt
Isotropic Sky
Hay & Davies (circumsolar, isotropic diffuse)
Reindl (circumsolar, isotropic diffuse, horizon brightening)
Perez (circumsolar, isotropic diffuse, horizon brightening)

The horizontal radiation models used are correlations estimating the diffuse fraction of total horizontal radiation. All use as a correlation variable some function of the ratio of total radiation on the horizontal surface to extraterrestrial radiation, k_T , also called the clearness index, to estimate the beam or diffuse component. The component of radiation which is not the object of correlation is then found subtractively. The full Reindl correlation is not used since the relative humidity, ϕ , for the testing periods is unknown. The parameters used in the various models are provided in Table 6.1, where α_{sa} is the solar altitude angle, T_a is the ambient temperature and Sc is the solar constant.

All of the tilt radiation models use the same technique for predicting the portion of the horizontal beam radiation that is incident on the tilt by the use of the geometric factor R_b , given in eqn (6.7). The portion of radiation on a tilted surface that is reflected from the ground is determined by assuming that the ground acts as an isotropic reflector. Thus, R_{gnd} is given in eqn (6.8), where $(1 - \cos\gamma)$ is the view factor from the collector to the ground, γ being the collector tilt angle.

$$G_{bT} = G_{bH} \cdot R_b, \quad R_b = \cos\theta / \cos\theta_z \quad (6.7)$$

$$G_{gndT} = G_H \cdot R_{gnd}, \quad R_{gnd} = 0.5(1 - \cos\gamma)\rho_g \quad (6.8)$$

The difference between the tilt radiation models comes from how the sky diffuse radiation is projected onto the tilt. The isotropic sky model assumes that the diffuse radiation is uniformly distributed. This model is not considered here since it underpredicts total tilt radiation and is not recommended for general use [TRNSYS, 1994]. The Hay and Davies model accounts for both circumsolar diffuse (an increased intensity of diffuse radi-

ation around the sun) and isotropic diffuse by defining a portion of the diffuse radiation to be circumsolar by the use of an anisotropy index. The Reindl model adds a horizon brightening diffuse term by adding it to the isotropic diffuse term and controlling the magnitude by a modulating factor. The Perez model includes all three terms through empirically derived "reduced brightness coefficients." The diffuse terms considered by each model are listed in Table 6.2 beneath the models' name.

6.3.2 Tau Alpha Product

The $\tau\alpha$ product, $(\tau\alpha)$, is now described. The $(\tau\alpha)$ is a transmittance-absorptance term for the cover and absorber plate of a flat-plate collector. τ is the transmittance for an incident beam through a single layer of glass, and α is the absorptance of the absorber plate. α is assumed to be constant, independent of incidence angle.

τ is a product of two transmittance terms, a reflection term that allows for reflection losses of unpolarized light and an absorption term that allows for absorption losses. The reflection term is a function of both the incident angle, θ , and the refracted angle, β ; both angles are shown in Fig. 6.1. β and θ are related through Snell's Law, given in Fig. 6.1. The absorption term is a function of path length through the glass, $L/\cos\beta$, and the extinction coefficient. These two terms are given in Duffie and Beckman [1991].

The transmittance-absorptance product, $(\tau\alpha)$, is not just the product of τ and α . Whether it be the beam, the diffuse, or the ground reflected component, $(\tau\alpha)$ is a term that accounts for the infinite number of reflections and corresponding absorption by the absorber plate of an incident beam of radiation transmitted through a cover. The reflections from the absorber plate are assumed to be diffuse and unpolarized.

Figure 6.1 Beam Refraction

The $(\tau\alpha)$ is dependent on the angle of incidence; therefore to calculate the total radiation transmitted through the cover and absorbed at the absorber plate, the $(\tau\alpha)$ would need to be integrated over all angles.

The average transmittance absorptance product, $(\tau\alpha)_{\text{ave}}$ is a ratio of the total absorbed radiation at the absorber plate to the solar radiation incident on the tilt. This average includes all three components, beam, sky diffuse, and ground reflected. On a clear sunny day when the beam fraction is high, $(\tau\alpha)_{\text{ave}}$ will be approximately equal to $(\tau\alpha)_b$, but on a cloudy day $(\tau\alpha)_{\text{ave}}$ may actually be closer to $(\tau\alpha)_{\text{sky}}$.

Since portions of sky diffuse and ground reflected incident components of radiation may be considered isotropic [Duffie and Beckman, 1991], the total transmitted radiation for each may be calculated by integrating over all angles. Therefore, $(\tau\alpha)_{\text{sky}}$ and $(\tau\alpha)_{\text{gnd}}$ may be defined for equivalent incidence angles. This may be simplified by assigning an equivalent beam angle that gives the same transmittance as for the sky diffuse radiation, and a different equivalent beam angle that gives the same transmittance as for the ground reflected radiation. These angles will be dependent upon the view factor and hence

the collector slope, γ . Average equivalent angles for both $(\tau\alpha)_{\text{gnd}}$ and $(\tau\alpha)_{\text{sky}}$ are given in eqn's (6.9) and (6.10), respectively. For a 45° collector tilt, $\theta_{\text{gnd}}=56.5^\circ$ and $\theta_{\text{sky}}=69.4^\circ$.

$$\theta_{\text{gnd}} = 90 - 0.5788\gamma + 0.002693\gamma^2 \quad (6.9)$$

$$\theta_{\text{sky}} = 59.7 - 0.1388\gamma + 0.001497\gamma^2 \quad (6.10)$$

6.3.3 $F_R U_L$

The second most important term in the basic efficiency equation, the heat loss factor, $F_R U_L$, is now examined. The heat loss coefficient is a function of the operating conditions of the collector, the ambient conditions, and the material properties of the collector. Therefore, U_L is affected by many continuously varying factors, even possibly the incidence angle (as the incidence angle increases, the incident solar decreases and the absorber plate temperature changes) [Thomas et al., 1982]. The collector heat removal factor, F_R , is given in eqn (6.11) and the overall loss coefficient, U_L , is given in eqn (6.12). The collector efficiency factor, F' , and the fin efficiency factor, F (which is embedded in F') are given in Appendix D for reference. The derivation of all these equations is provided in Duffie and Beckman [1991].

$$F_R = B \left[1 - e^{-\left(\frac{F}{B}\right)} \right], \quad B = \frac{\dot{m} C_p}{A_c U_L} \quad (6.11)$$

$$U_L = U_{\text{top}} + U_{\text{back}} + U_{\text{edge}} = \left[\frac{1}{h_w + h_{r_{c-a}}} + \frac{1}{h_{c_{p-c}} + h_{r_{p-c}}} \right]^{-1} + \frac{k_{\text{ins}}}{L_{\text{ins}}} \quad (6.12)$$

Where:

k_{ins} = insulation conductivity ($W/m^{\circ}C$)

L_{ins} = insulation thickness (m)

h_w = convective heat transfer coefficient due to wind ($W/m^2^{\circ}C$)

h_r = radiative heat transfer coefficient ($W/m^2^{\circ}C$)

h_c = convective heat transfer coefficient ($W/m^2^{\circ}C$)

c-a = denotes cover to ambient

p-c = denotes plate to cover

The overall heat loss coefficient, U_L , is deeply embedded in the collector heat removal factor, F_R ; directly and through the collector efficiency factor, F' , and the fin efficiency factor, F . F_R is also dependent on the convective heat transfer coefficient for fluid flow inside the tubes, h_{fluid} . The flow is laminar, and so h_{fluid} is calculated using the Seider-Tate relation; $h_{fluid}=165 W/m^2^{\circ}C$.

U_L is difficult to determine exactly because of its dependence on many variables. The edge losses are considered to be negligible. The back losses are k_{ins}/L_{ins} . The wind convective coefficient, h_w , is dependent upon the wind speed, the convective coefficient between the glass and cover is dependent upon the Raleigh number (depends on the difference between the cover and plate temperatures) and the collector slope, and the radiative heat transfer coefficients are dependent upon differences of the driving temperatures to the fourth power.

6.4 Approach

Here, linearized models for collector performance are posited. A linear least squares program is used to obtain best-fit values of the parameters. Then these best-fit parameters are used back in the model to resimulate performance. The quality of the

parameters is assessed on a physical level and a visual level by the fit to the data of the resimulated results.

The modeling of the collectors is performed systematically; first by successively extending the absorbed radiation term through the use of different components of radiation and $(\tau\alpha)$. Second by extending the heat loss coefficient term, and last by combining the first two approaches. Each of the models results in at least two parameters to be determined from data. These parameters are evaluated by the method of linear least squares.

6.4.1 Absorbed Radiation

The absorbed radiation, S , is defined three ways as stated in section 6.4. The first, as defined in the HWB equation, combines the total incident radiation on the tilt with $(\tau\alpha)_{ave}$. This is termed the TT (Total Tilt) model and is given as:

$$S_{TT} = G_T(\tau\alpha)_{ave} \quad (6.13)$$

The second definition utilizes G_{bT} and G_{diffT} , the outputs from the TRNSYS radiation processor when using both a horizontal radiation model and a tilt model, with $(\tau\alpha)_b$ and $(\tau\alpha)_{diff}$. This model is termed the BD (beam and diffuse) model. S is given as:

$$S_{BD} = G_{bT}(\tau\alpha)_b + G_{diffT}(\tau\alpha)_{diff} \quad (6.14)$$

The final definition is comprised of all three radiation components and $(\tau\alpha)$ components. The outputs from the TRNSYS radiation processor, using a horizontal radiation

model only, G_{bH} and G_{skyH} , are used as input to the tilt radiation model. This model will be called BSDG (beam, sky diffuse, and ground reflected), and is given as:

$$S_{BSDG} = G_{bH}R_b(\tau\alpha)_b + G_{skyH}R_{sky}(\tau\alpha)_{sky} + G_H R_{gnd}(\tau\alpha)_{gnd} \quad (6.15)$$

The model used for R_{sky} , the Reindl model, is presented in Appendix D.

6.4.2 Tau Alpha Correlations

As stated in section 6.3.2, the $(\tau\alpha)$ product is dependent on the incidence angle. Therefore, in order to add fidelity to the model, and also to follow the SRCC procedure (given in section 6.2), IAM's are defined.

These correlations for the $(\tau\alpha)$ dependence on the incident direction are given in this section; the incident direction is denoted by θ for $(\tau\alpha)_{ave}$ and for $(\tau\alpha)_b$, by θ_{sky} for $(\tau\alpha)_{sky}$, and θ_{gnd} for $(\tau\alpha)_{gnd}$.

The most frequently used correlation for $(\tau\alpha)$ is from Souka and Safwat [1966]. If $(\tau\alpha)$ is plotted against $\cos \theta$, (θ is the incident angle) a curve such as is shown in Fig. 6.2(a) is obtained. If instead $(\tau\alpha)$ is plotted against $1/\cos\theta$ (shown in Fig. 6.2(b)), a linear relationship of $(\tau\alpha)$ versus θ holds for values of θ less than 68° . At larger angles the behavior begins to deviate and the curve actually goes to infinity at 90° . The non-linearity after 68° is not extremely important, since only a small fraction of the overall daily energy is incident at angles greater than 68° . The systems' pumps may not even turn on at the times of day that these incident angles occur, but the energy will pre-heat the stagnant fluid in the collector.

Figure 6.2 (a) $(\tau\alpha)$ Angle Relationship; (b) $(\tau\alpha)$ 1/Angle Relationship

Since the $(\tau\alpha)$ at normal incidence, $(\tau\alpha)_n$, is known from manufacturers data, the curve fit given in eqn (6.16) is a ratio of $(\tau\alpha)$ to $(\tau\alpha)_n$, which is called the incident angle modifier (IAM).

$$IAM_{\theta 1} = \frac{(\tau\alpha)}{(\tau\alpha)_n} = \left[1 - b_0 \left(\frac{1}{\cos\theta} - 1 \right) \right] \quad (6.16)$$

In an attempt for even greater accuracy, the linear relationship for IAM can be extended to a second order relationship:

$$IAM_{\theta 2} = \left[1 - b_0 \left(\frac{1}{\cos\theta} - 1 \right) - b_1 \left(\frac{1}{\cos\theta} - 1 \right)^2 \right] \quad (6.17)$$

Since $(\tau\alpha)$ is actually a function of both β and θ , the behavior of the IAM with β and $1/\cos\beta$ may also be used. This behavior is also plotted in Fig. 6.2(a) and(b). Note that the curves appear similar to those using θ . Thus, the above two correlations are used with the refracted angle β in place of θ , eqn's (6.18) and (6.19). These two correlations would be good only for $(\tau\alpha)_{ave}$ or $(\tau\alpha)_b$, but not for $(\tau\alpha)_{diff}$ and $(\tau\alpha)_{gnd}$; the equivalent angles β_{diff} (37.8°) and β_{gnd} (33.1°) are calculated from θ_{diff} and θ_{gnd} . The correlations for θ are good only up to 68° angle of incidence, which corresponds to a maximum 37.5° angle of refraction, β .

$$IAM_{\beta 1} = \left[1 - b_0 \left(\frac{1}{\cos\beta} - 1 \right) \right] \quad (6.18)$$

$$IAM_{\beta 2} = \left[1 - b_0 \left(\frac{1}{\cos\beta} - 1 \right) - b_1 \left(\frac{1}{\cos\beta} - 1 \right)^2 \right] \quad (6.19)$$

With modern computing capabilities, though, it is not necessary to compromise on the accuracy of the equation to fit $(\tau\alpha)$ versus θ . One of the problems with using $1/\cos\theta$ is that, at large angles, the IAM value tends to negative infinity as θ goes to 90°. An approach

which does not suffer from this unphysical behavior is to fit a polynomial curve to $(\tau\alpha)$ versus $\cos\theta$ or $\cos\beta$. A fourth order polynomial in $\cos\theta$, eqn (6.20), is used.

$$IAM_{poly} = (1 + b_0\cos\theta + b_1(\cos\theta)^2 + b_2(\cos\theta)^3 + b_3(\cos\theta)^4) \quad (6.20)$$

The IAM must be equal to 1 at θ equal to 90° . Thus, $b_0+b_1+b_2+b_3 = 0$.

6.4.3 Heat Loss Models

As noted in section 6.2, Klein [1973], asserts that the transient effect is due to fluctuations in the wind speed, not fluctuations in the radiation intensity. This sentiment is also expressed by Gupta et al. [1978], and Bowen [1980]. This may be corroborated by examining relative magnitudes in the $F_R U_L$ term.

Each of the convective and radiative heat transfer coefficients is estimated for low and high typical data values for the natural convection heat exchanger system (see Appendix D). The wind speed varies between 1 and 10 m/s; h_w is calculated from the correlation given in Duffie and Beckman [1991] and varies from 5 to 18 $W/m^2\text{ }^\circ C$. h_r for cover to ambient (the ambient temperature is actually T_{sky} , which is assumed to be $10^\circ C$ less than the ambient temperature [Duffie and Beckman, 1991]) varies between 3 and 6 $W/m^2\text{ }^\circ C$, and h_r for the plate to the cover varies from 3.5 to 8 $W/m^2\text{ }^\circ C$. h_c for the plate to the cover varies approximately from 3 to 4 $W/m^2\text{ }^\circ C$. All of these terms seem to have equal importance in regards to relative values, but as can be inferred the most variable is the convective coefficient due to wind speed. Since the wind speed is constantly changing, it will indeed have the largest impact on the variability of the overall heat loss coefficient. h_{wind} also has

an inherent sensitivity to the wind direction and the turbulence in the fluid stream, but overall these sensitivities are low [Kind et al., 1983; Green, 1987]. The greatest effect is due to variations in wind speed from 0 to 1.2 m/s [Green, 1987].

The extreme complexity of the $F_R U_L$ term would necessarily lead to non-linear least squares. In order to keep the parameters linear, the $F_R U_L$ term is divided into a linear function of wind speed, v_{wnd} as given in eqn (6.21). The powers on the wind speed, n , are chosen to be 1 and also 0.6, as recommended in the correlation given in Duffie and Beckman [1991].

$$F_R U_L = a v_{wnd}^n + b \quad (6.21)$$

6.4.4 Parametric Equations

The above formulations for S may be incorporated into eqn (6.2) or eqn (6.3) to yield the collector performance equations. The number of parameters to be determined will be dependent on the IAM correlation used, but there will always be at least three: F_R , $F_R b_0$, and $F_R U_L$.

Next, S is taken as S_{TT} , no $(\tau\alpha)$ correlations are used, and the formulations for $F_R U_L$ are used to yield the collector performance equations with unknown parameters $F_R(\tau\alpha)_{ave}$, 'a', and 'b'.

Finally, the formulations for both S and $F_R U_L$ are used in eqn (6.2) or eqn (6.3) to yield the collector performance equations.

The performance equations are initially developed using the collector efficiency, eqn (6.3), for easier comparison to the values used by the SRCC, and to be consistent with the ASRHAE testing procedure.

6.4.4.1 Performance Equations in S

Using the various formulations for S, which includes $(\tau\alpha)$'s (equal to $IAM^*(\tau\alpha)_n$), 16 performance equations may be developed. Examples for each radiation model, S, are given with $IAM_{\theta 1}$. The remaining combinations of radiative models with the IAM correlations are presented in Table 6.3 along with the parameters to be determined. Using the TT radiative model, with no $(\tau\alpha)_{ave}$ correlations, results in eqn (6.22), with the two parameters $F_R(\tau\alpha)_{ave}$ and $F_R U_L$ to be determined.

$$\begin{aligned}\eta_{col} &= F_R \frac{S_{TT}}{G_T} - F_R U_L \left(\frac{T_{in, col} - T_a}{G_T} \right) \\ &= F_R (\tau\alpha)_{ave} - F_R U_L \left(\frac{T_{in, col} - T_a}{G_T} \right)\end{aligned}\tag{6.22}$$

This is a convenient equation in that a minimum of test data, three temperature measurements, one incident radiation measurement, and one collector flow rate measurement only are needed to determine the performance parameters, $F_R(\tau\alpha)_{ave}$, and $F_R U_L$. Once the data are obtained, the collector efficiency may be plotted versus $(T_{in, col} - T_a)/G_T$, and "best-fit" to a straight line. $F_R(\tau\alpha)_{ave}$ is the intercept, and $F_R U_L$ is the slope. The linear least squares routine described in section 6.4.5 is used to determine these two parameters.

If $(\tau\alpha)_{ave}$ in the TT model is expanded using eqn (6.16) for $(\tau\alpha)$, the result is eqn (6.23). The parameters to be determined are F_R , $F_R b_0$, and $F_R U_L$; manufacturer's data are

used to determine $(\tau\alpha)_n$, and experimental data are used for η_{col} , and $(T_{in,col}-T_a)/G_T$; θ is computed from the solar angle relations.

$$\eta_{col} = F_R(\tau\alpha)_n - F_R b_0(\tau\alpha)_n \left(\frac{1}{\cos\theta} - 1 \right) - F_R U_L \left(\frac{T_{in,col} - T_a}{G_T} \right) \quad (6.23)$$

If the $(\tau\alpha)$ terms in the BD model are expanded using eqn (6.16), eqn (6.24) is the result. The model for IAM versus θ is used for both beam and diffuse, except the known equivalent incident angle for sky diffuse, θ_{diff} , is used in $(\tau\alpha)_{diff}$. The unknown parameters are F_R , $F_R b_0$, and $F_R U_L$.

$$\eta_{col} = F_R(\tau\alpha)_n - F_R b_0 \frac{(\tau\alpha)_n}{G_T} \left\{ G_{bT} \left(\frac{1}{\cos\theta} - 1 \right) + G_{dT} \left(\frac{1}{\cos\theta_{diff}} - 1 \right) \right\} - F_R U_L \left(\frac{T_{in,col} - T_a}{G_T} \right) \quad (6.24)$$

The same procedure is used with the BSDG model and eqn (6.16) to develop eqn (6.25). The equivalent angles for θ_{diff} and θ_{gnd} are again used in $(\tau\alpha)_{diff}$ and $(\tau\alpha)_{gnd}$. The unknown parameters are F_R , $F_R b_0$, and $F_R U_L$. Experimental data are used to determine the remaining variables in the equation; these are then used in the linear least squares program.

$$\eta_{col} = F_R(\tau\alpha)_n - F_R b_0 \frac{(\tau\alpha)_n}{G_T} \left[G_{bH} R_b \left(\frac{1}{\cos\theta} - 1 \right) + G_{skyH} R_{sky} \left(\frac{1}{\cos\theta_{sky}} - 1 \right) + G_{gndH} R_{gnd} \left(\frac{1}{\cos\theta_{gnd}} - 1 \right) \right] - F_R U_L \left(\frac{T_{in,col} - T_a}{G_T} \right) \quad (6.25)$$

Table 6.3 Summary of Optics Models

Models	Unknown Parameters					
S_{TT} eqn (6.13)	$F_R(\tau\alpha)_{ave}$	F_{RU_L}				
S_{TT} eqn (6.13) $IAM_{\theta 1}$ eqn (6.16)	F_R	F_{Rb_0}	F_{RU_L}			
S_{TT} eqn (6.13) $IAM_{\theta 2}$ eqn (6.17)	F_R	F_{Rb_0}	F_{Rb_1}	F_{RU_L}		
S_{TT} eqn (6.13) $IAM_{\beta 1}$ eqn (6.18)	F_R	F_{Rb_0}	F_{RU_L}			
S_{TT} eqn (6.13) $IAM_{\beta 2}$ eqn (6.19)	F_R	F_{Rb_0}	F_{Rb_1}	F_{RU_L}		
S_{TT} eqn (6.13) IAM_{poly} eqn (6.20)	F_R	F_{Rb_0}	F_{Rb_1}	F_{Rb_2}	F_{Rb_3}	F_{RU_L}
S_{BD} eqn (6.14) $IAM_{\theta 1}$ eqn (6.16)	F_R	F_{Rb_0}	F_{RU_L}			
S_{BD} eqn (6.14) $IAM_{\theta 2}$ eqn (6.17)	F_R	F_{Rb_0}	F_{Rb_1}	F_{RU_L}		
S_{BD} eqn (6.14) $IAM_{\beta 1}$ eqn (6.18)	F_R	F_{Rb_0}	F_{RU_L}			
S_{BD} eqn (6.14) $IAM_{\beta 2}$ eqn (6.19)	F_R	F_{Rb_0}	F_{Rb_1}	F_{RU_L}		
S_{BD} eqn (6.14) IAM_{poly} eqn (6.20)	F_R	F_{Rb_0}	F_{Rb_1}	F_{Rb_2}	F_{Rb_3}	F_{RU_L}
S_{BSDG} eqn (6.15) $IAM_{\theta 1}$ eqn (6.16)	F_R	F_{Rb_0}	F_{RU_L}			
S_{BSDG} eqn (6.15) $IAM_{\theta 2}$ eqn (6.17)	F_R	F_{Rb_0}	F_{Rb_1}	F_{RU_L}		
S_{BSDG} eqn (6.15) $IAM_{\beta 1}$ eqn (6.18)	F_R	F_{Rb_0}	F_{RU_L}			
S_{BSDG} eqn (6.15) $IAM_{\beta 2}$ eqn (6.19)	F_R	F_{Rb_0}	F_{Rb_1}	F_{RU_L}		
S_{BSDG} eqn (6.15) IAM_{poly} eqn (6.20)	F_R	F_{Rb_0}	F_{Rb_1}	F_{Rb_2}	F_{Rb_3}	F_{RU_L}

6.4.4.2 Model for $F_R U_L$

The term in the overall heat loss coefficient that has the possibility of being the most variable is due to the convective wind losses. Therefore, the $F_R U_L$ term is formed into a linear combination of the wind speed, together with a constant, b , which encompasses natural convective and radiative losses.

$$\eta_{col} = F_R(\tau\alpha)_{ave} - (av_{wnd}^n + b)\left(\frac{T_{in,col} - T_a}{G_T}\right) \quad (6.26)$$

6.4.4.3 Combined Models

Finally, the models for efficiency summarized in section 6.3.6 for the optics models are used in conjunction with the wind speed models of section 6.3.7 to provide even greater physical resolution. The combined models may have up to seven parameters to be determined; F_R , $F_R b_0$, $F_R b_1$, $F_R b_2$, $F_R b_3$, a , and b .

6.4.5 Least Squares

A linear least squares method is used to determine the parameters in the equations. Linear least squares is a means to fit a set of data (x_i, y_i) to a model which is a linear combination of any P specified functions of x , $X(x)$, as given in eqn (6.27) [Numerical Recipes, 1989].

$$y(x) = a_0 + a_1\chi_1 + a_2\chi_2 + \dots + a_P\chi_P = \sum_{j=0}^P a_j\chi_j \quad (6.27)$$

The X_j functions can be any function of x , either simple polynomials or complicated combinations including sines and cosines; these are called the basis functions. The parameters, a_j , must be linear. The merit function is defined as the sum of the squares of the deviations at each data point i , $(y_i - \sum a_j x_{ij})$. The derivative of the merit function with respect to all P parameters yields P equations, called the normal equations, which effect a minimum by being set to zero.

The recommended technique to be used for the solution of these equations is singular value decomposition. This is used because, in many cases, the normal equations are very close to singular. This may occur when data do not clearly distinguish between two or more basis functions. If any singular value is zero, its reciprocal is also set to zero, rather than infinity. This corresponds to adding to the fitted parameters a zero multiple, not some large multiple, of any linear combination of basis functions which are degenerate in the fit. The program that is used for the least squares fits is based on a program from *Numerical Recipes* [1989].

Also included in the program are statistical analyses of the parametric equations to determine the covariance and cross-correlation matrices [Draper and Smith, 1966; Wadsworth, 1990]. The program is constructed specifically for this problem, and is included on the disk in the back of the dissertation.

6.4.6 Quality of Fit Metrics

One criterion for the selection of the models is based on the chi-squared distribution. The chi-squared distribution is the distribution that would be followed by squared

normal random variables; hence the chi-squared is a measure of "goodness of fit" of the variance. The chi-squared is given as:

$$\chi^2 = \frac{\sum_{i=1}^M (y_i - y_{pred,i})^2}{M} \quad (6.28)$$

A second criteria used is the r^2 which is a measure of the value of the regression equation. It is the percentage of the variability in y attributed to using the fitted parameters and X functions to predict y :

$$r^2 = \frac{\sum_{i=1}^M (y_{pred,i} - y_{mean})^2}{\sum_{i=1}^M (y_i - y_{mean})^2} \quad (6.29)$$

6.4.6.1 Radiation Processor

To determine which horizontal model and which tilt model to use to process the experimental data, the TRNSYS radiation processor is run for all 12 possible combinations of horizontal and tilt models. The runs use a data set containing five days of 5 minute averaged total horizontal radiation data. The total tilt radiation output from these runs is then compared to the total measured tilt radiation. The combination of horizontal and tilt models which best-fit the measured data, based on chi-squared and r^2 as given above, are determined. The best-fit horizontal and tilt models are used thereafter for all the data sets. In the process of exercising the models, an error was discovered in one of the horizontal models, as is described in the results, section 6.5.1.1.

6.4.6.2 Collector Performance Equations

The best-fit models are chosen based on chi-squared, r^2 , and also the most analytically correct parameters. Once the best models are determined, they are modified in two ways. First, fits are also considered using a constant $F_R U_L$ as determined from night time operation of the system (this removes $F_R U_L$ as a parameter during the processing of the day time data), as described below. The parameters determined for the best models using the collector efficiency equation are then re-determined using the collector useful energy equation, eqn (6.2).

The parameters $F_R(\tau\alpha)_{ave}$, and $F_R U_L$, determined experimentally, are then compared to the parameters used in the SRCC ratings. The efficiencies and useful energies predicted using the experimentally determined parameters and the SRCC parameters are compared to the measured efficiencies and useful energies to determine if/what additional parameters (for the collector optics and heat loss) are needed to the basic model to improve the prediction of the collector performance.

6.4.7 Possible Runs

The experimental data used to determine the parameters described above are from the continuous operation tests. The collector parameters determined from the least squares program are determined using the day time data, when there is incident radiation, so that there is both an optics component and a losses component. The program may also be run with any percentage of maximum incident radiation values desired; this essentially centers

the data points around solar noon. This is used to calculate $F_R(\tau\alpha)_{ave}$ and $F_R U_L$ from clear day data. These values may be directly compared to the SRCC values.

6.4.8 Night Time Data Analysis

A constant $F_R U_L$ is determined from the night time data. If the system is forced to operate at night, an energy balance on the system results in eqn (6.31), given as an average. Two methods were considered for averaging $F_R U_L$, given as eqn (6.30) and eqn (6.31).

$$\overline{F_R U_L} = \frac{\dot{m} C_p (\bar{T}_{out, col} - \bar{T}_{in, col})}{A_c (\bar{T}_{in, col} - \bar{T}_a)} \quad (6.30)$$

$$\overline{F_R U_L} = \frac{\dot{m} C_p (T_{out, col} - T_{in, col})}{A_c (T_{in, col} - T_a)} \quad (6.31)$$

Although each method predicts an $F_R U_L$ of the same order, the second method is considered to be more accurate since it averages individual energy calculations, rather than averaging measurements and then calculating the energy. Method 1 would assist in removing noise. Therefore, the average $F_R U_L$ is calculated at each time step, and then the average is calculated from all measurements:

$$\frac{\dot{m}_i C_p (T_{out, col, i} - T_{in, col, i})}{A_c (T_{in, col, i} - T_{a, i})} = -(F_R U_L)_i \quad (6.32)$$

$$(F_R U_L)_{ave} = \frac{\sum_{i=1}^M (F_R U_L)_i}{M}$$

Note that F_R is independent of θ , so this value should not change appreciably during the day, except perhaps due to a higher ambient temperature, or substantially different wind speeds. $F_R U_L$, though, could change appreciably with ambient conditions. The variation of F_R and $F_R U_L$ with plate temperature is shown in Fig. 6.3. The day time curves are

Figure 6.3 F_R and $F_R U_L$ Variation with Plate Temperature

for an average ambient temperature of 15°C and an average plate temperature of 55°C (determined as the average of the collector inlet plus the collector outlet temperatures) and the night time curves are for an average ambient temperature of 0°C and an average plate

temperature of 2°C. The values for F_R and $F_R U_L$ are calculated as described in Duffie and Beckman [1991], page 261, with $h_{wind} = 7 \text{ W/m}^2\text{C}$, $\epsilon_{plate} = 0.95$, and a collector slope of 45°. This variation in $F_R U_L$ will be addressed in the results.

The best-fit parameters chosen are then used with another data set to predict the efficiencies and useful energies measured during a different test. The predicted values for the second data set are compared against the actual values.

6.5 Results

Presented in this section are the results of using the collector models presented above which were “best-fit” to the data. As the most critical input to the collector is the solar radiation, an in-depth analysis of the radiation models is presented first. This provides insight for estimation of collector parameters, which is presented thereafter.

6.5.1 Radiation Processor

The twelve possible combinations of horizontal and tilt models, shown in Table 6.1, are processed for the data set January 12-17, 1996. The results for three of the combinations are given, the poorest best-fit, the “best-fit”, and the model chosen. The combination with the best-fit uses the reduced Reindl horizontal model, but this model is not considered further for reasons given below.

The results for all three model combinations are presented in Table 6.4. The predicted beam radiation on the tilt for January 13 (the clear day) is similar for all of the models; the difference is greatest in the prediction of the diffuse radiation on the tilt. All of the

models underpredict the total radiation on the tilt on this day. On January 14 (the cloudy day) the Boes and Hay and Davies models predict higher beam radiation than the other two models and higher diffuse than the Erbs and Perez models, and overpredicts the daily total radiation on the tilt. This overprediction is consistent with the observations of Fanney and Liu [1980]. Included in the table is the r^2 , the standard error (square root of the chi-squared), and the average radiation value for all of the days, for the three radiation models discussed here. The r^2 of all the models are similar. Basing the goodness of fit on the standard error and the average, the Erbs horizontal model with the Perez tilt model is better than the Boes horizontal model with the Hay and Davies tilt model.

Table 6.4 Predicted Components of Tilt Radiation

Model	Component	Total	Jan 13	Jan 14	r^2	S.E.	Y_{ave}
Measured	Total Tilt	83.7	24.1	18.3	-	-	0.170
Boes Horiz Hay & Davies Tilt MJ/m ²	Predicted Beam	67.5	19.1	14.4	0.91	0.018	0.177
	Predicted Diffuse	16.1	4.1	3.8			
	Predicted Total Tilt	85.8	23.97	18.9			
Erbs Horiz Perez Tilt MJ/m ²	Predicted Beam	66.3	19.2	13.9	0.88	0.013	0.172
	Predicted Diffuse	13.6	3.2	3.7			
	Predicted Total Tilt	82.8	23.1	18.2			
Reindl Horiz Reindl Tilt MJ/m ²	Predicted Beam	62.7	18.9	12.6	0.88	0.011	0.173
	Predicted Diffuse	19.97	3.9	4.9			
	Predicted Total Tilt	83.4	23.6	18.2			

The combination of models with the poorest best-fit is the Boes horizontal model with the Hay & Davies tilt model. The tilt radiation, both measured and computed in TRN-SYS, and the predicted beam and diffuse components on the tilt are shown in Fig. 6.4 for January 13 and 14, 1996. January 13 was a very clear day, and January 14 was a semi-

cloudy day. On the clear day, the tilt radiation is overpredicted in the early morning and late evening and underpredicted around noon, the tilt is predicted correctly for one to two hours on either side of noon. On the cloudy day the tilt radiation is overpredicted overall, but both overpredicted and underpredicted throughout the day. The largest overpredictions occur when there are large sharp increases in radiation level (a cloud moves out from in front of the sun), and the largest underpredictions occur when there are large rapid decreases in radiation level (a cloud moves in front of the sun).

Figure 6.4 Boes Horizontal and Hay & Davies Tilt Radiation Models

The Erbs horizontal model with the Perez tilt model are chosen as the models to use for all of the data sets. The measured and predicted tilt radiations for January 13 and

14 for these models are shown in Fig. 6.5. These models underpredict the tilt radiation on the clear day and slightly on the cloudy day.

Figure 6.5 Erbs Horizontal and Perez Tilt Radiation Models

6.5.1.1 Reindl Horizontal Model

A plot of the measured tilt radiation and beam and diffuse components computed in TRNSYS using the Reindl horizontal and Reindl tilt modes is shown in Fig. 6.6 for January 13 and 14 (representative days). Note the precipitous drop in beam/rise in diffuse around noon.

Figure 6.6 Reindl Horizontal and Reindl Tilt Models

Reasons for the unphysical behavior are now explored. The reduced Reindl horizontal radiation model uses both k_T and the solar altitude angle, α_{sa} to predict the diffuse component of radiation on the horizontal. The equations used are:

$$\text{Interval: } 0 \leq k_T \leq 0.3; \quad \text{Constraint: } I_{dH}/I_H \leq 1.0$$

$$I_{dH}/I_H = 1.020 - 0.254k_T + 0.0123 \sin \alpha_{sa}$$

$$\text{Interval: } 0.3 < k_T < 0.78; \quad \text{Constraint: } 0.1 \leq I_{dH}/I_H \leq 0.97$$

$$I_{dH}/I_H = 1.400 - 1.749k_T + 0.177 \sin \alpha_{sa}$$

$$\text{Interval: } 0.78 \leq k_T; \quad \text{Constraint: } I_{dH}/I_H \geq 0.1$$

$$I_{dH}/I_H = 0.486k_T - 0.182 \sin \alpha_{sa}$$

(6.33)

If I_{dH}/I_H is plotted against k_T , the curve should appear as shown in Fig. 6.7. As the clearness index increases, then the fraction of diffuse should decrease to a minimum. At very low values of the clearness index (very cloudy), then most of the incident radiation is diffuse; at large values of the clearness index (very clear), then most of the incident radiation is beam. Figure 6.7 also shows I_{dH}/I_H versus k_T for the data collected for January 12-17, 1996. Notice the sharp discontinuity in slope at $k_T = 0.78$. Problems arise, especially in winter in Colorado (or regions of very sunny winter days), when k_T exceeds 0.78 and the solar altitude does not get much larger than 30° , as may occur in winter throughout the day, but is most noticeable near solar noon. Due to this erroneous behavior, this model is not considered further.

Figure 6.7 Diffuse Fraction versus Clearness Index

6.5.1.2 Discussion

Each of the models produces a large spike, labeled S on figures 6.4-6.6, at the beginning of each day. This occurs because very early in the morning, the horizontal model underpredicts the sky diffuse on the horizontal. As a result, the beam radiation is overpredicted. When this large value of beam radiation is multiplied by the very large value of R_b for this time, a gross overprediction of the beam on the tilted surface results. This is not a concern when running the least squares program since there is very little incident solar flux available at this time of day. These spikes appear only at isolated instances, and are independent of calculations at other times.

Accurately predicting from horizontal data the components of radiation incident on a tilted surface is a difficult task. The radiation models need to be general and based on values relatively easily measured, therefore it is unreasonable to expect a model to work perfectly. The problem of predicting the components of incident radiation is compounded in Colorado where it is possible to have "bright" cloudy days and to get higher radiation readings than on clear days (due to reflections from the clouds). This is possible even in the winter.

The methods of predicting the diffuse and beam components of incident radiation has implications when performing parameter estimation. All optics models, except for the one using measured data, use the separate components of radiation as inputs. A greater ratio of diffuse radiation to total radiation leads to a higher IAM since this necessarily means lower beam radiation which lowers $(\tau\alpha)_b$; $(\tau\alpha)_{diff}$ varies to a lesser degree since it is dependent on $1/\cos\theta_{diff}$, which is constant. Conversely, a greater ratio of beam radiation to total radiation leads to a lower IAM since $(\tau\alpha)_b$ is dependent on $1/\cos\theta$, which is vari-

able. This may be one problem with using solar simulator tests to determine collector performance parameters, even though such tests are easier to perform. A solar simulator has the lowest ratio of diffuse to total irradiance and should give the lowest IAM curve [Thomas et al., 1982].

6.5.2 Parameter Estimation

The results of the parameter estimation are presented in this section. All results are obtained using data collected from continuous operation tests. First, all optical parameters and the heat loss parameter are determined by performing a least squares fit using the day-time data. From these results an absorbed radiation model is chosen to be used in the subsequent parameter estimation. Second, the performance collector equation is given as based on either efficiency or useful energy. Third, the results are presented for determining a constant $F_R U_L$ from night-time data. Fourth, the collector optical parameters are determined from the day-time data using the constant $F_R U_L$ determined from the night-time data, adjusted to account for increased day-time temperatures. Fifth, the heat loss parameter, $F_R U_L$, is linearized with wind speed. Finally, the predicted parameters are compared with the SRCC parameters and the predicted parameters and the SRCC parameters are used to predict the collector performance for a second set of data.

6.5.2.1 Determination of All Parameters

Parameter estimation is performed as described in earlier sections for the collector of the natural convection heat exchanger system. The results are presented in Table 6.5 for

Table 6.5 Collector Parameter Estimates

Correlation	$F_R(\tau\alpha)_n$	b_0	b_1	b_2	b_3	$F_R(\tau\alpha)$	$F_R U_L$	r^2	S.E.	Q_{use}
Measured	-	-	-	-	-	$\theta = 30^\circ$	$\frac{W}{m^2 \cdot ^\circ C}$	-	-	204.5 MJ
S_{TT} eqn (6.13)	0.434	-	-	-	-	0.434	1.440	0.29	0.102	191.3
S_{TT} eqn (6.13) $IAM_{\theta 1}$ eqn (6.16)	0.489	0.390	-	-	-	0.459	1.088	0.64	0.072	199.6
S_{TT} eqn (6.13) $IAM_{\theta 2}$ eqn (6.17)	0.490	0.397	-0.005	-	-	0.460	1.083	0.64	0.072	199.6
S_{TT} eqn (6.13) $IAM_{\beta 1}$ eqn (6.18)	0.514	2.037	-	-	-	0.572	0.929	0.62	0.075	200.1
S_{TT} eqn (6.13) $IAM_{\beta 2}$ eqn (6.19)	0.479	0.067	8.572	-	-	0.463	1.030	0.64	0.073	199.6
S_{TT} eqn (6.13) IAM_{poly} eqn (6.20)	-2.1	-6.18	12.63	-11.79	4.108	0.464	1.086	0.65	0.072	199.7
S_{BD} eqn (6.14) $IAM_{\theta 1}$ eqn (6.16)	0.530	0.391	-	-	-	0.498	0.164	0.76	0.059	204.9
S_{BD} eqn (6.14) $IAM_{\theta 2}$ eqn (6.17)	0.533	0.426	-0.021	-	-	0.498	0.174	0.71	0.059	204.9
S_{BD} eqn (6.14) $IAM_{\beta 1}$ eqn (6.18)	0.562	2.311	-	-	-	0.486	0.369	0.71	0.065	204.5
S_{BD} eqn (6.14) $IAM_{\beta 2}$ eqn (6.19)	0.503	-0.91	13.10	-	-	0.453	0.200	0.76	0.059	204.7
S_{BD} eqn (6.14) IAM_{poly} eqn (6.20)	0.387	-8.37	28.77	-31.65	11.60	0.502	0.176	0.77	0.059	204.8
S_{BSDG} eqn (6.15) $IAM_{\theta 1}$ eqn (6.16)	0.489	0.452	-	-	-	0.455	-0.206	0.89	0.062	205.0
S_{BSDG} eqn (6.15) $IAM_{\theta 2}$ eqn (6.17)	0.486	0.424	0.016	-	-	0.454	-0.202	0.91	0.062	204.9
S_{BSDG} eqn (6.15) $IAM_{\beta 1}$ eqn (6.18)	0.537	2.789	-	-	-	0.45	-0.203	0.91	0.073	204.6
S_{BSDG} eqn (6.15) $IAM_{\beta 2}$ eqn (6.19)	0.456	-1.27	16.13	-	-	0.397	-0.192	0.91	0.062	204.7
S_{BSDG} eqn (6.15) IAM_{poly} eqn (6.20)	-1.097	-5.25	8.362	-6.764	2.223	0.457	-0.120	0.89	0.062	204.8

every combination of collector equation and correlations for $(\tau\alpha)$. The collector performance equation is based on efficiency, eqn (6.3). The first column ("correlation") presents the model used for S , and the equation number for that model. The second row of the first column is the equation number of the $(\tau\alpha)$ correlation used. The " $F_R(\tau\alpha)$ " column gives $F_R(\tau\alpha)$, as calculated with the $F_R(\tau\alpha)_n$ ($(\tau\alpha)_n$ is 0.864 as given by the manufacturer) and b_0 parameters, at an incidence angle of 30° . The "S.E." column is the standard error in the efficiency (the square root of the chi-squared value). For comparison, Proctor [Part 2, 1984] states that with current instruments the best obtainable variance (error) is ± 0.02 in absolute efficiency for normal incidence and ± 0.028 in absolute efficiency for off-normal incidence. Most of this error is attributed to the pyranometer. The " Q_{use} " column is the total predicted five day useful energy (MJ), calculated as the sum of the predicted efficiency (from the estimated parameters) times the collector area times the incident radiation. Note that the actual measured value is 204.5 MJ.

One problem, though, is the determination of $F_R U_L$. For the S_{BD} model, $F_R U_L$ is on the order of 0.2, and for the S_{BSDG} model, $F_R U_L$ is actually even negative. These values are much lower than the value of $3 \text{ W/m}^2\text{C}$, calculated directly from night time data. One reason for this is that the correlation between the χ functions of the F_R and $F_R U_L$ terms is high (0.6 or greater); therefore as $F_R(\tau\alpha)$ decreases, $F_R U_L$ also decreases. This is not unexpected; as the incidence angle increases from zero at normal incidence, the incident radiation is necessarily decreasing. As the incident radiation decreases, the absorber plate temperature decreases and hence so does $F_R U_L$.

Thus, these low predicted values of $F_R U_L$ also suggest that the $F_R(\tau\alpha)$ values are also low. This could easily occur if the predicted beam component of radiation is low and

the predicted diffuse component of radiation is high. Therefore, if a model were to be chosen based on the $F_R U_L$ closest to the experimental value, it would be the S_{TT} model. Noting all of this, it becomes difficult to choose a model to use for S .

For all of the models for S , all the $(\tau\alpha)$ correlations give very similar results. The $(\tau\alpha)$ correlation using a fourth-order cosine function, eqn (6.18), is marginally better overall than the other $(\tau\alpha)$ correlations. But the added complexity may not warrant using it; especially since there is a very high statistical correlation between the variables, giving very large variances for each parameter; and importantly, note that $F_R(\tau\alpha)_n$ is not physically correct (negative) for the S_{TT} and S_{BSDG} models.

Given all of the above observations, all models for S (S_{TT} , S_{BD} , and S_{BSDG}) using a linear $(\tau\alpha)$ correlation in $1/\cos\theta$, eqn (6.16), are chosen for further study. Figure 6.8 shows the predicted useful energies (defined as $\eta^* A_c G_T$) for models S_{TT} , S_{BD} , and S_{BSDG} with eqn (6.16), as well as the measured useful energy, for January 13 and 14, 1996. Notice on January 13 the noon time draw is clearly shown for the measured useful energy. These draws are not discerned by the parameter estimation since there are so many fewer data points collected during draws than collected overall. A high weighting factor would need to be used during the draw data, to yield parameters which capture the detail of the draws.

6.5.2.2 Efficiency and Useful Energy Collector Equation

Two approaches are used in the determination of the collector performance parameters. First, the parameters are determined using the collector efficiency as the performance equation, eqn (6.3). This is consistent with the ASHRAE procedure; $F_R(\tau\alpha)_n$ and

Figure 6.8 Predicted Useful Energy ($\eta \cdot A_c G_T$) Over Time for All Parameters Predicted

$F_R U_L$ are determined as the intercept and slope of the collector efficiency versus $(T_{in,col} - T_a)/G_T$. Ultimately, though, the collector useful energy is the quantity of interest. This may be calculated as $\eta_{col} A_c G_T$.

Second, the parameters are determined using the collector useful energy as the performance equation, eqn (6.2). The weighting of the data is now greater at noon and less in the early morning and late afternoon, since the χ functions no longer include the incident radiation in the denominator. This is important since the greatest influence on the $(\tau\alpha)$ terms should indeed be at noon.

If the parameter identification is performed based on useful energy, eqn (6.2), as opposed to efficiency, eqn (6.3), an improved prediction is obtained for all models. The

improvement is realized because the artificial weighting of the early morning and late afternoon data is removed. These results are given in Table 6.6 and in Fig 6.9 for the useful energy using model S_{TT} with the $(\tau\alpha)$ correlation of eqn. 6.16.

Figure 6.9 Predicted Useful Energy Using Efficiency and Useful Energy Collector Performance Equations

6.5.2.3 Determination of $F_R U_L$ from Night-Time Data

A constant average $F_R U_L$ is calculated from the night-time data as described in section 6.5.7, eqn (6.32). For the natural convection heat exchanger system, the instantaneous $F_R U_L$'s had to be calculated using only the first one to two hours of night-time data. Because there is not a large volume of glycol, the glycol temperature quickly cools down to the ambient temperature. Once these two temperatures are approximately equal, the

Table 6.6 Parameters for Efficiency and Useful Energy Collector Performance Equations

Model	eqn	Parameter			S.E.	Q _{use} (MJ)
		F _R (τ α) _n	b ₀	F _R U _L		
Measured	—	-	-	-	-	204.5
S _{TT} (τ α) _{ave}	η	0.43	-	1.44	0.10	191.3
	Q _u	0.5	-	2.48	229	208.3
S _{TT} eqn (6.16)	η	0.49	0.39	1.09	.07	199.6
	Q _u	0.50	0.36	1.20	146	205.2
S _{BD} eqn (6.16)	η	0.53	0.39	0.16	0.06	204.9
	Q _u	0.54	0.39	-0.29	152	204.4
S _{BSDG} eqn (6.16)	η	0.49	0.46	-0.21	0.06	205.0
	Q _u	0.48	0.47	-0.7	156	203.6

“noise” in the data tends to overwhelm the signal, thereby negatively affecting the ability to compute $F_R U_L$ accurately. The data collected from the first night are also not used since the test is started with all temperatures equal. The instantaneous $F_R U_L$'s, along with collector inlet and outlet, and ambient temperatures, are shown in Fig 6.10 for January 13 and 14, 1996. Notice how quickly the temperatures approach each other at night. The very sharp decrease in $T_{in,col}$ during the day occurs when a draw is performed; cold water from mains supply goes to the bottom of the solar tank, which circulates through the heat exchanger, thus cooling the glycol.

The calculated night-time average $F_R U_L$, for the natural convection heat exchanger system, is 3 W/m²°C. The $F_R U_L$ determined analytically (using the method given in section 6.4.8) is 2.7 W/m²°C (for night-time plate and ambient temperatures) to 3.4 W/m²°C (for day-time plate and ambient temperatures). The experimentally determined average

Figure 6.10 Instantaneous $F_R U_L$

value of $3 \text{ W/m}^2\text{°C}$ is higher than the analytical night-time value of $2.7 \text{ W/m}^2\text{°C}$ due to increased losses from the collector manifold (equivalent to two small fins).

The average value of $F_R U_L$ computed from the night-time data is taken as known, and increased by 20% to account for the higher day-time plate and ambient temperatures. The day-time data are then used to compute an “instantaneous” value of $F_R(\tau\alpha)$ at each time step:

$$F_R(\tau\alpha)_i = \eta_{col,i} + (F_R U_L)_{ave} \left[\frac{(T_{in,col} - T_a)}{G_T} \right]_i \quad (6.34)$$

The “instantaneous” $F_R(\tau\alpha)$ ’s, as well as the incident radiation, and $(T_{in,col} - T_a)/G_T$, are shown in Fig. 6.11. This plot shows the variability in $F_R(\tau\alpha)$ throughout the day. At low efficiencies, the $(T_{in,col} - T_a)/G_T$ term in eqn (6.34) is “high,” while at high efficiencies the $(T_{in,col} - T_a)/G_T$ term is “low.” Therefore one would expect $F_R(\tau\alpha)$ to remain relatively constant. This does not happen, though, because the overall variability in the $(T_{in,col} - T_a)/G_T$ term is much less than the variability in the efficiency, thus, $F_R(\tau\alpha)$ tends to follow the efficiency curve. An average $F_R(\tau\alpha)$ for the day-time data is 0.47.

Figure 6.11 Instantaneous Parameter Variation

6.5.2.4 Determination of Optical Parameters with Known $F_R U_L$

If the $F_R U_L$ parameter is forced to be equal to the average value calculated from night-time data, the quality of the least squares fit decreases. However, the decrease is not especially significant, and the parameters are more realistic. $F_R(\tau\alpha)$ remains between 0.5 and 0.55 for almost all of the models. Also, there is only a 3% difference in $F_R(\tau\alpha)$ between using $F_R U_L$ equal to 3.0 and $F_R U_L$ equal to 3.6. These results are summarized in Table 6.7. Figures 6.12 ($F_R U_L=3$) and 6.13 ($F_R U_L=3.6$) show the energy, using the collector performance equation based on useful energy, for models S_{TT} and S_{BD} with eqn (6.16).

6.5.2.5 $F_R U_L$ as a Linear Function of Wind Speed

The correlations for $F_R U_L$ with wind speed, Table 6.8, give results that are approximately equivalent to the results using the S_{TT} model with no $(\tau\alpha)$ correlations. This is because the 5-minute averaged wind speed does not vary much over the data set, as apparent in Fig. 6.14. The average wind speed is 1.7 m/s, which when used with the parameters a and b , yields an $F_R U_L$ of about 1.5 W/m²°C. Kind et al. [1983] state that a linear relationship for h_{wind} with wind speed does not account for changes in the Reynolds number nor for the turbulence in natural wind. This needs to be investigated further. Similar models have been proposed by Proctor [Part 2, 1984] and Gillett [1980] except that they include a radiation term. In order to keep the model linear, the sky temperature is needed. Since the sky temperature was not measured here, this should be studied in future work.

Figure 6.12 S_{TT} and S_{BD} , $FRUL=3$

Figure 6.13 S_{TT} and S_{BD} , $FRUL=3.6$

Table 6.7 Parameters for $F_R U_L = 3$ and 3.6

Model	eqn	Parameter			S.E.	Q_{use} (MJ)
		$F_R(\tau\alpha)_n$	b_0	$F_R U_L$		
Measured	-	-	-	-	-	204.5
S_{TT} $(\tau\alpha)_{ave}$ $F_R U_L = 3$	η	0.5	-	3.0	.10	201.3
	Q_u	0.51	-	3.0	230	207.3
S_{TT} $(\tau\alpha)_{ave}$ $F_R U_L = 3.6$	η	0.45	-	3.6	.14	205.1
	Q_u	0.45	-	3.6	236	206.1
S_{TT} eqn (6.16) $F_R U_L = 3$	η	0.54	0.25	3.0	0.11	209.1
	Q_u	0.54	0.27	3.0	168	202.6
S_{TT} eqn (6.16) $F_R U_L = 3.6$	η	0.57	0.22	3.6	0.13	212.2
	Q_u	0.55	0.25	3.6	184	201.7
S_{BD} eqn (6.16) $F_R U_L = 3$	η	0.54	0.11	3.0	0.12	207.3
	Q_u	0.57	0.21	3.0	201	202
S_{BD} eqn (6.16) $F_R U_L = 3.6$	η	0.54	0.05	3.6	0.14	207.8
	Q_u	0.58	0.17	3.6	218	201.5
S_{BSDG} eqn (6.16) $F_R U_L = 3$	η	0.54	0.24	3.0	0.15	209.6
	Q_u	0.54	0.36	3.0	247	196.4
S_{BSDG} eqn (6.16) $F_R U_L = 3.6$	η	0.56	0.21	3.6	0.18	210.4
	Q_u	0.56	0.35	3.6	271	195.2

Table 6.8 Results with $F_R U_L$ a Linear Function of Wind Speed

Correlation	$F_R(\tau\alpha)$	a	b	$F_R U_L$ ($v=1.7$ m/s)	r^2	S.E.	Q_{use}
measured	-	J/m ² °C	W/ m ² °C	W/m ² °C	-	-	204.5 MJ
$F_R U_L$ $av+b$	0.432	-0.560	2.486	1.534	0.314	0.100	191.8
$F_R U_L$ $av^{0.6}+b$	0.432	-1.275	3.278	1.525	0.314	0.100	191.7

Figure 6.14 Wind Speed Variation

6.5.3 Comparison of Predictions

Since all of the models predict very similar results, it is recommended that the simplest model be used. The results presented in this section are based on the S_{TT} model, linear in $1/\cos\theta$ (eqn (6.16)). The collector performance is based on the useful energy (eqn (6.3)) instead of efficiency. The constant $F_R U_L$ is the night-time average increased by 20%, ($3.6 \text{ W/m}^2\text{°C}$).

First, the results are compared to the SRCC predicted performance. Second, the predicted performance parameters are used to predict the performance for a second data set.

6.5.3.1 Comparison to SRCC

The useful energies predicted using parameters estimated from in-situ testing are compared against the useful energies predicted using the collector parameters determined through the SRCC [Solar Rating & Certification Corporation, 1994] ratings program. The SRCC testing follows the ASHRAE [American Society for Heating, Refrigerating, and Air Conditioning Engineers, 1988] protocol.

The ASHRAE test procedure requires that the incident radiation be greater than 750 W/m^2 , with diffuse radiation accounting for less than 20%. The ambient must be between 15 and 30°C, but not varying more than 1.5°C for the duration of the test. Four data points are required at each of the four values of $(T_{\text{in,col}} - T_a)$ specified; specifically, $T_{\text{in,col}}$ is to be 0, 30, 60, and 90% of the manufacturers allowed maximum fluid temperature (200°C for the natural convection heat exchanger system). Two of the four data points must be taken before, but near, solar noon, and the other two are required at the symmetric point after solar noon. The points are integrated over 5 minutes. This gives a total of 16 data points spanning 80 minutes. The collector parameters determined for the natural convection heat exchanger system as given by the SRCC [Solar Rating & Certification Corporation, 1994] are: $F_R(\tau\alpha)_n=0.637$, $F_R U_L=4.54 \text{ W/m}^2\text{°C}$, and $b_0 = 0.1559$.

This test with high incident radiation and low diffuse radiation simulates data taken around solar noon on clear days. This equation works reasonably well when the objective is predicting the annual useful energy gained by the collectors since the largest amount of energy gained will indeed be around solar noon on clear days. But if one were interested in predicting the performance of the collector at other times of the day, under

different solar conditions, or even in predicting the temperature rise across the collector, this relation would not prove as useful.

An attempt was made to obtain 16 data points from the test data that best corresponds to the ASHRAE requirements. Around solar noon, it would be necessary to obtain $(T_{in,col} - T_a)$ equal to 0°C , 55°C , 111°C , and 166.5°C for the natural convection heat exchanger system. From the data it was possible to obtain a range of only 10 to 25°C . This is reasonable since neither $T_{in,col}$ nor T_a are controlled. Using the 16 points over this range, a linear fit resulted in $F_R(\tau\alpha)_n$ equal to 0.56 and $F_R U_L$ equal to $5.2 \text{ W/m}^2\text{C}$ with an r^2 of 0.41. If the slope of the line is changed to $F_R U_L = 5.2 \text{ W/m}^2\text{C}$, a 30% change, then $F_R(\tau\alpha)_n$ becomes 0.53, a change of only 5%, with an r^2 of 0.35. Both of these lines fit the data almost equally well. $F_R(\tau\alpha)_n$ predicted by all of the methods are within 10% of each other giving confidence in the value, yet this variability in $F_R(\tau\alpha)_n$ can lead to as much as 40% variability in $F_R U_L$.

Under continuous operation test conditions, Fig. 6.15 shows $F_R(\tau\alpha)_n$ predicted for various percentages of the available data. The horizontal axis is the percentage of the data used to calculate $F_R(\tau\alpha)_n$, centered around solar noon. For example, 10% is 5% of the data on either side of solar noon. 100% uses all of the day time data. The SRCC value is also shown. The minimum percentage that yields a good statistical fit for $F_R(\tau\alpha)_n$ is 10%. 10% of the day time data for the duration of a 5 day test is about 6 hours of 5 minute averaged data points. This is many more data points than required by the ASHRAE test procedure.

Notice how all methods converge to $F_R(\tau\alpha)_n$ equal to 0.53 to 0.55 if only "sunny" solar data are used. This is possible if 10% to 35% of the data is used. At larger cumulative percentages of data, the diffuse nature of the incident radiation comes more into play, as

does the off-normal incidence angle modifier. As more diffuse data are added, the spread in $F_R(\tau\alpha)_n$ becomes larger. This is due to the high correlation between $F_R(\tau\alpha)_n$, $F_R U_L$, and b_0 . This is also shown by the large variability in $F_R(\tau\alpha)_n$ when both $F_R(\tau\alpha)_n$ and $F_R U_L$ are predicted through parameter estimation (S_{TT} -Eff-Var). The SRCC predicted value of $F_R(\tau\alpha)_n$ equal to 0.637 is also shown. This is much larger than the value of 0.55 predicted using experimental data and parameter estimation. One cause for this difference may be the weatherization of the collectors. The collectors used in this test have been operated under normal operating conditions for about a year and are more than likely dirtier than the collectors used in the SRCC testing program. The SRCC value for $F_R U_L$ is 4.54 W/m²°C. This is also much higher than the value of 3.6 predicted from the night time data. The SRCC incidence angle modifier term, b_0 , is 0.156 and is lower than the estimated parameter, using the data, of 0.25.

Figure 6.16 shows the useful energy predicted using the SRCC parameters, the useful energy predicted using the S_{TT} model, $F_R U_L = 3.6$ W/m²°C, and based on the useful energy performance equation. The useful energy calculated from measured data is also shown. The SRCC model consistently overpredicts the useful energy. This is as expected since the SRCC predicted optics parameter (the dominant term during this period of days with high incident radiation), $F_R(\tau\alpha)_n$, is 15% larger than the experimentally determined parameter while the less influential loss parameter, $F_R U_L$, is only 20% larger. As stated previously, a 30% change in $F_R U_L$ required only a 5% change in $F_R(\tau\alpha)_n$ to maintain essentially the same curve fit for noon time data.

Figure 6.15 $F_R(\tau\alpha)_n$ vs. Percentage of Incident Radiation

6.5.3.2 Comparison to Second Data Set

The data collected during another continuous operation test, November 14-17, 1996, are input into the parameter estimation program. The new parameters, for the S_{TT} model, using eqn (6.16) and based on useful energy for the performance equation, are given in Table 6.9, along with the parameters determined previously using the January 12-17, 1996 data set. The Q_{useful} column is the total predicted useful energy for November 14-17 using the given parameters. The average $F_R U_L$ calculated for the November 14-17

Figure 6.16 Predicted Useful Energy using SRCC Parameters

night-time data is $3.1 \text{ W/m}^2\text{°C}$, which is within 3% of the average $F_R U_L$ determined using the January 12-17 data. This is as expected and gives confidence in the estimated value.

Table 6.9 Predicted Useful Energy for November 14 - 17, 1996

Model	Test Dates Parameters Derived From:	Parameter			Q_{useful} MJ
		$F_R(\tau\alpha)_n$	b_0	$F_R U_L$	
Measured	—	—	—	—	132.9
S_{TT} eqn (6.16) Q_u	Jan. 12 -17	0.50	0.36	1.20	138.4
	Nov. 14-17	0.51	.33	1.81	130.7
S_{TT} eqn (6.16) Q_u $F_R U_L=3.6$	Jan. 12 -17	0.55	0.25	3.6	130.4
	Nov. 14-17	0.55	0.24	3.6	128.2
SRCC	—	0.637	0.156	4.54	117.2

The predicted useful energy for the November test, using the parameters determined from the January test is presented in Fig.6.17. The predicted useful energy using the SRCC parameters is also shown. The performance parameters determined from the January test data successfully predict the useful energy for both data sets; the January useful energy is underpredicted by 1%, and the November data is overpredicted by 4%. The SRCC parameters overpredict the January useful energy by 17% and underpredict the November useful energy by 12%.

Figure 6.17 Predicted Useful Energy, November 14-17, 1996

The incident radiation for the November data set includes two clear days and one extremely cloudy day. The daytime hours are longer than for the January data set, and the

clear days are less cloudy in the early morning and late evening than for the January data set. The peak incident radiation is about the same for both data sets. The collector inlet temperature is higher for the November data set, and the ambient temperature is about the same, or slightly lower for the November data set. Therefore, the temperature difference in the collector loss term, $(T_{in,col} - T_a)$, increases by 10 to 15°C. This increases the significance of the loss term, thus explaining the behavior using the SRCC parameters (a large $F_R U_L$ parameter). This also suggests that $F_R U_L$ determined from the data and parameter estimation may be slightly low.

6.6 Summary

The performance of a flat-plate collector may be modeled as having two terms: an optics term, $F_R(\tau\alpha)_n$, which accounts for the absorption of incident radiation, and a loss term, $F_R U_L$, which accounts for the heat transfer between the collector and the environment. The optics term of the collector is the most important, accounting for as much as 80% of the energy gain.

The optics term of the collector was analyzed in-depth through separation of $(\tau\alpha)$ into beam, diffuse, and sky reflected components. The added complexity in the model does not contribute significant statistical improvement in the fit; especially if the 'goodness' of the models used to determine the beam, diffuse, and sky reflected components of the incident radiation are considered.

Initially, the optics and collector loss parameters were both determined using a parameter estimation program. This yielded unphysical parameters because of the high correlations between the parameters. Thus, it is suggested that the collector loss parameter

be determined from night time test data. This night-time $F_R U_L$ should then be increased to account for the increased day-time operating temperatures. The optics parameters, $F_R(\tau\alpha)_n$ (for direct incident radiation), and b_0 (the incident angle modifier), may then be determined using parameter estimation techniques. It was determined that at least 6 hours of 5 minute averaged data are needed to yield a good statistical fit to predict $F_R(\tau\alpha)_n$ under actual operating conditions.

The parameters estimated from the data in this manner predict the collector energy to within 1% for the data set used to obtain the parameters. These same parameters, when used with a different data set, predict the useful energy to within 4%. Therefore, this is judged to be an acceptable method for determining the parameters to predict the performance of a flat-plate collector.

The parameters given by the SRCC for this collector overpredicts the useful energy by 17% on data set 1, and underpredicts the useful energy by 12% on data set 2. Thus, an overall annual average is acceptable, but better prediction results from parameter estimation in the method outlined in this dissertation.

CHAPTER 7 CONCLUSIONS AND RECOMMENDATIONS

7.1 Introduction

This chapter presents the major conclusions of the work, and concludes with recommendations for future work.

7.2 Conclusions

The conclusions that result from this work may be categorized as either general, therefore applicable to all systems, or system specific. Some relate to the design of systems, and some to the impact on the SRCC ratings procedure.

7.2.1 General

To determine the parameters of a solar domestic hot water system, data should be collected under realistic conditions. Thus, the instrumentation of the systems has been implemented with great care and detail. The resulting energy balances on the systems are good to within 10%, thus giving confidence in the data.

The tank heat losses in systems may be as high as 10 to 15% of the useful energy, therefore, calculating them correctly is important. The calculated UA of a hot water tank under the SRCC program is determined by filling a stand-alone tank with hot water and measuring the temperature decay over time. If the test is performed with the tank plumbed

into the system, the losses increase by as much as 25% over the stand-alone tank UA value because of the significant heat loss from the piping. Installing heat traps into the inlet and outlet lines of the tank can decrease the losses by 10 to 15%.

Analysis of in-situ data for the flat-plate collectors shows that the optics term of the collector is the most important, accounting for as much as 80% of the energy. Though this may be the case, it was determined that using a more complicated model (such as using more parameters by separating the optics term, $(\tau\alpha)_{ave}$, into different components corresponding to the different components of incident radiation) is not worthwhile. This is due to the high correlation between the parameters and also to the inexactness of the radiation processors.

The estimation of the collector loss coefficient and the optical parameters together leads to a high statistical correlation between parameters, thus yielding, at times, unphysical parameters. This may be removed as an issue if the loss coefficient is determined separately from the experimental data. If the system is operated over night, then there will be no gains at the collector and the losses may be calculated. However, the loss coefficient needs to be increased (by about 20%) to account for the difference in ambient and collector plate temperatures between the night time operation and day time operation.

The parameter estimation program used in this dissertation provided parameters capable of estimating the collector useful energy within 1% on the data set used to obtain the parameters, and to within 4% for a different data set. The 'correct' $F_R(\tau\alpha)_n$ was determined with a minimum of 8 hours of high incident radiation ('noon') data.

The parameters recommended by the SRCC ratings program overpredict the useful energy gain on one data set by 17% and underpredict on the second data set by 12%, thus

overall, the annual prediction may be reasonable. The $F_R(\tau\alpha)_n$ is 14% greater than that estimated from data, and $F_R U_L$ is 21% greater. The SRCC parameters are determined from 16 data points for a total of 80 minutes. The difference between the estimated parameters and the SRCC estimated parameters may be attributed, in part, to the fact that these tests were performed with seasoned collectors, not new collectors.

7.2.2 System Specific

There is significant heat loss, as much as 30%, due to backflow through the thermosyphon loop of the natural convection heat exchanger system. The backflow may occur whenever the pump is not operating and the glycol in the heat exchanger is cooler than the storage tank's hot water. In order to improve the performance, a very low pressure check valve could be installed in the thermosyphon loop to prevent this backflow.

The analysis of the load side heat exchanger in the drain-back system resulted in a more complete understanding of the system's operation. It was determined that the outside convective coefficient of the heat exchanger is the dominant resistance and that the performance of the heat exchanger is best predicted using a horizontal cylinder correlation with a 20 to 25% enhancement due to local turbulence generated from flow over the closely stacked coils. As a conclusion of this analysis, an improvement to increase the system's performance may be suggested. The present design circulates supply water, at a high flow rate, through the main solar storage tank's heat exchanger, thus extracting the collected solar energy only during a draw. If a continuous flow of fluid is allowed to circulate through the coil heat exchanger and into the auxiliary tank, a greater percentage of the

energy stored in the main solar storage tank may be removed. A design to accomplish this has been implemented and tested [Arent, 1996].

The modeling of the load-side heat exchanger aids the SRCC efforts at rating such systems. The proposed model, which uses an average heat exchanger effectiveness to model the heat exchanger, is deemed acceptable for systems which perform "shallow" energy draws. TRNSYS simulations were performed to correspond to the experimental test dates. TRNSYS predicted the useful energy to within 1% and the draw energy to within 10%.

The ICS system was not discussed in detail, but TRNSYS simulations were performed and reported by Smith et al. [1995]. The simulations yielded differences from the actual measured data by about 25%, leading to the conclusion that the TRNSYS Type 46 model needs to be improved. The possible problems include determination of the IAM's and the UA of the "collectors." Since back plane reflectance becomes important with the ICS system, the collector parameters need to be determined with dependence on more variables, such as back plane reflectance and tube spacing. The UA of the collectors may be different when water is flowing through the tubes than that when it is not because of interaction (heat exchange) between the flow streams. The UA values used in the TRNSYS model were determined for no flow in the tubes. Also, the collector modules should be plumbed in parallel, not series; in series 2/3 of the total useful energy gained was in the first module [Smith et al., 1995].

7.3 Recommendations

The conclusions generated from this research lead directly to recommendations that can be made to improve the performance of the systems, changes to ratings testing procedure, and future work that should be performed.

The storage tanks of hot water systems should be tested with all plumbing installed when determining the tank loss coefficients to obtain a more realistic value. To improve the UA value, heat traps should be installed in the plumbing lines entering/leaving the tanks. A low pressure check valve should be installed in natural convection heat exchanger systems to decrease the heat losses due to backflow. The flow circulation patterns that result in the piping entering and leaving the tanks should be explored further to achieve greater understanding of the methods of heat loss from the tanks. This can be extended to include studying the tank losses with and without flow in the systems.

The performance of the drain-back system with the load-side heat exchanger can be improved by redesigning the system so that there is continual natural convection flow between the auxiliary tank and the main solar storage tank. In this manner more of the collected solar energy will be extracted. The TRNSYS model for the load-side heat exchanger using a constant effectiveness yields satisfactory results, but the model could be improved still further. This has been accomplished, but the specific model has not been verified in this dissertation.

The testing and modeling of the collectors resulted in many observations relevant to the SRCC ratings program. The determination of the collector parameters $F_R(\tau\alpha)_n$, b_0 , and F_RU_L from the experimental data exhibited a high statistical correlation. Thus, it is necessary to determine one of the parameters in another manner. This is accomplished by

calculating $F_R U_L$ by operating the system at night while measuring the inlet and outlet temperatures of the fluid flowing through the collector, as well as the flow rate through the collector. To be more accurate, the night sky radiation should also be measured. Another method would be to install in-line heaters in the collector loop. To obtain a statistically significant value of $F_R(\tau\alpha)_n$ at least 8 hours of 5 minute averaged data are required. The model should be modified to allow parameters resulting from allowing the absorptance term to vary with incidence angle. This is especially important for collectors with selective surfaces.

The reason for the significant difference between the experimentally determined collector parameters and the collector parameters determined using the SRCC ratings procedure should be investigated. Two possible reasons for these differences are because this work uses seasoned collectors and that this work averages both clear and cloudy weather data, not just clear sky data.

TRNSYS runs should be performed for the collectors using both the SRCC parameters and the experimental parameters. First, though, the radiation models provided in TRNSYS should be further analyzed. To do this, both the beam and diffuse components of radiation should be measured for these tests. They should also be performed at during each season of the year. Once the radiation models are improved it may be possible to improve the collector optical parameter, $F_R(\tau\alpha)_n$, by including dependence on the different radiation components (beam, diffuse, and sky diffuse).

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APPENDIX A

Instrumentation List

Pyranometers

Eppley PSP

Temperature dependence: $\pm 1\%$ over ambient range -20 to $40\text{ }^{\circ}\text{C}$

Linearity: $\pm 0.5\%$ from 0 - 2800 W/m^2

Response time: 1 second

Cosine: $\pm 1\%$ from normalization $0 - 70^{\circ}$ zenith angle

(1) S/N 16726F3, Tilt

(2) S/N 16696F3, Horiz.

Wind Speed/Direction

Climatronics WM-III

P/N 100108

$6.4\text{ V} \pm 0.25\text{ mV}$ supply (6 mA)

Digi-Key Power-One International Series DC Power Supply

Model HAD15-0.4-A

Regulation: $\pm 0.05\%$

Burr-Brown Frequency to Voltage Converter

VFC 320

Linearity: $\pm 0.005\%$ @ FS

Watt Hour Meters

GE Watt-hour meter 60 Hz

Type IW-70-5

Cat No. 720x72G115

(1) S/N 67 666 716 Accuracy: $\pm 0.4\%$

(2) S/N 67 666 713 Accuracy: $\pm 0.55\%$

(3) S/N 67 666 739 Accuracy: $\pm 0.5\%$

Watt Transducers

Ohio Semitronics (for Auxiliary Heaters)

(1) PC-5-50CX5	(2) PC5-29	(3) PC5-59
200-280V, 0-50 A	240V, 50A	240V, 100A
1 phase, 2 wire	1 phase, 2 wire	1 phase, 2 wire
current ratio 50:5, 2 turns	current transducer, 2 turns	current transf., 1 turn
S/N 21691	S/N 3024	S/N 3451
Accuracy: +/- 0.5% at Rated	+/- 0.5% at Rated	+/- 0.5% at Rated
ICS	Nat. Conv. HX	Drain-back

F. W. Bell Inc. (for pumps)

(1) PX-2202B	(2) PX-2202B	(3) PX-2204B
10 A/20 A, 120 V, 60 Hz	10 A/20 A, 120 V, 60 Hz	1 A/2 A, 120 V, 60 Hz
S/N 107492	S/N 101500	S/N 124724
Accuracy: +/- 0.5%	Accuracy: +/- 0.5%	Accuracy: +/- 0.5%
ICS	Drain-back	Nat. Conv. HX

Flow Meters

Flow Technologies Inc.

FTO-4N1YW-LHC-1	FT-12NEXS-LAD-2
Normal Flow Range: 0.07-0.7 gpm	Normal Flow Range: 2 - 20 gpm
Extended Flow Range: 0.02-1.3 gpm	Extended Flow Range: 0.4 - 25 gpm
Accuracy: +/- 0.05%	Accuracy: 0.05%
< 10 psid at normal 10:1 max flow rate	< 10 psid at normal 10:1 max flow rate
(1) Drain-back S/N F088083075	(1) Supply S/N 1203908
(2) Nat. Conv. HX (coll.side) S/N F088083074	

Cox Flow Meters

Series 21 1/2 - 15
Normal Flow Range: 1.5 - 15 gpm
Accuracy: +/- 0.5%
9.5 psid @ mac flow rate
(1) ICS S/N 25943
(2) Drain S/N 44157

Micromotion Type UL-D-IS

DS0655239SU S/N 165552
Normal Flow Range: 0 - 10 lb/min
Accuracy: +/- 0.05%
(1) Nat. Conv. HX (Tank side)

Flow Meter Boxes

FTI, LN51-1-C-I000-6, 0-10kHz, 0-5V DC output, 115V AC input
use with FTO-4
S/N LN5188080180

S/N LN5188080179

FTL RC51-1-C-I000-6

S/N RC5189030109, 0-5V out, (FT-12)

Cox 1000-P, ASS'Y No. 107-051

S/N 5-0096M-23546

S/N 5-0093M-23554

Range Cap = 233 pf

Range Cap = 222 pf

1200 Hz = 35.2 gpm @ 1.174V, 115VAC 1200Hz=34.3 bpm @ 1.142V, 115VAC

Micromotion Transmitter, RFT97121PNUR, S/N 49354

Pressure Transducers

Elxon Power Systems Power Supply Supply OLV15-12, 12 V, 1.5 A

Omega Engineering

PX26-015 DV

PX26-030 DV

Range: 0-15 psid

Range: 0-30 psid

Accuracy: +/- 1% Full Scale

Accuracy: +/- 1% Full Scale

(1) Drain-back

Nat. Conv. HX

(2)ICS

HP75000 PC Data Acquisition System 10

	<u>Resolution</u>	<u>System Accuracy</u>
<u>DC Voltage:</u> 3V	7.6e-6V	0.01%+54e-6 V
30 V	61e-6 V	0.015%+1e-3 V

SOCOS 386 Computer

T - Type Thermocouples

8 for each Tank = 40

2 for each collector = 6

Drain-back = 4

ICS = 4

Nat. Conv. HX = 5

1 indoor ambient

1 outdoor ambient

1 supply

1 drain

Accuracy:

Absolute = +/- 0.1 °C

Differential = +/- 0.05 °C

Thermocouples

Accuracy: 0.5 °C

TOTAL = 63

APPENDIX B

Error Analysis

A detailed error analysis follows. Independent errors have been combined in quadrature, as recommended by J. P. Holman in "Mechanical Measurement and Instrumentation," McGraw Hill, 1988. The instrument errors are listed in Appendix A.

The data presented in Table B.1 are used as representative of the three domestic hot water solar systems on a typical day. Known values are used where possible. In the following derivations, the errors in specific heat and density associated with the temperature changes are considered negligible.

B.1 Determining the Error in $\dot{Q}_{\text{useful}}$, $F_R(\tau\alpha)$ and $F_R U_L$

An energy balance across the collector results in the following equation

$$\dot{Q}_{\text{useful}} = A_c [F_R(\tau\alpha)_{\text{ave}} G_T - F_R U_L (T_{\text{in, col}} - T_a)] = \dot{m} c_p (T_{\text{out, col}} - T_{\text{in, col}}) \quad (\text{B.1})$$

Using representative values for the drain-back system collector inlet and outlet temperatures from Table B.1, the useful energy is given as:

$$\dot{Q}_{\text{useful}} = \left(6.32 \times 10^{-5} \frac{\text{m}^3}{\text{s}}\right) \left(1000 \frac{\text{kg}}{\text{m}^3}\right) \left(4180 \frac{\text{J}}{\text{kg}^\circ\text{C}}\right) (10^\circ\text{C}) = 2642 \text{ W} \quad (\text{B.2})$$

The error in $\dot{Q}_{\text{useful}}$ is given as

Table B.1 Typical Property Values

Variable	Collector System	Value
Collector Inlet Temp. (°C)	Drain-Back	40
	ICS	40/22
	Nat. Conv. HX	40
Collector Outlet Temp. (°C)	Drain-Back	50
	ICS	50
	Nat. Conv. HX	90
$F_R(\tau\alpha)_n$	Drain-Back	0.48
	Nat. Conv. HX	0.55
$F_R U_L$ (W/m ² °C)	Drain-Back	3.6
	Nat. Conv. HX	4.0
Volume Flow Rates (m ³ /s)	Drain-Back	6.32×10^{-5}
	ICS	6.32×10^{-5}
	Nat. Conv. HX	2.08×10^{-5}
	Draw	2.0×10^{-4}
Pump Rating (W)	Drain-Back	29.8
	ICS	7.5
	Nat. Conv. HX	248.7
Pressure Head (kPa)	Drain-Back	47.3
	ICS	10.3
	Nat. Conv. HX	145
Specific Heat (J/kg°C)	Water	4180
	Glycol	3819
Density (kg/m ³)	Water	1000
	Glycol	1007
Outdoor Ambient Temp. (°C)	-	30/15
Indoor Ambient Temp. (°C)	-	24/19
Supply Temp. (°C)	-	21/17
Draw Temp. (°C)	-	55
Global Irradiance (W/m ² °C)	-	870/1025
Auxiliary Heater Rating (W)	-	3000

$$E_{Q_{useful}} = \sum \left[\left(E_V \frac{\partial Q}{\partial V} \right)^2 + \left(E_{\Delta T} \frac{\partial Q}{\partial \Delta T} \right)^2 \right]^{1/2} = \sum \rho c_p \dot{V} \Delta T \left[\left(\frac{E_V}{\dot{V}} \right)^2 + \left(\frac{E_{\Delta T}}{\Delta T} \right)^2 \right]^{1/2} \quad (B.3)$$

$$E_{Q_{useful}} = 2642 W \left((0.0005)^2 + \left(\frac{0.05}{10} \right)^2 \right)^{1/2} = 13.3 W = 0.5 \text{ percent} \quad (B.4)$$

The same percentage applies to the ICS system when the pump is operating. If a draw is being performed, the temperature difference increases since supply water, instead of the auxiliary tank water, enters the collector. The error percentage in Q_{useful} during a draw for the ICS system is $13.7 W = 0.2\%$. The error percentage in Q_{useful} for the natural convection heat exchanger system is $4.5 W = 0.1\%$.

The error in $F_R(\tau\alpha)$ and $F_R U_L$ is determined using the collector performance equation (first equality in eqn B.1) and the results for the error in Q_{useful} .

$$\begin{aligned} E_{Q_{useful}} &= 13.3 = [(G_T E_{F_R(\tau\alpha)})^2 + (F_R(\tau\alpha) E_{G_T})^2 + ((T_{in, col} - T_a) E_{F_R U_L})^2 + \\ &\quad (F_R U_L E_{T_{in, col}})^2 + (F_R U_L E_{T_a})^2]^{1/2} \\ &= [(870 E_{F_R(\tau\alpha)})^2 + (0.48(13.05))^2 + (10 E_{F_R U_L})^2 + 2(4(0.1))^2]^{1/2} \\ &\Rightarrow (870 E_{F_R(\tau\alpha)})^2 + (10 E_{F_R U_L})^2 = 13.3^2 - 39.2 - 0.32 = 137.3 \end{aligned} \quad (B.5)$$

The maximum error possible in $F_R(\tau\alpha)$ is if the error in $F_R U_L$ is assumed to be zero; and the maximum error in $F_R U_L$ occurs if the error in $F_R(\tau\alpha)$ is assumed to be zero. Therefore, for the drain-back system in the summer, the error in $F_R(\tau\alpha)$ is less than 0.013, or 2.8%, and the error in $F_R U_L$ is less than $1 W/m^2 \text{ } ^\circ C$, or 25%. (Drain-back system in winter: $F_R(\tau\alpha) < 0.011 = 2.2\%$, $F_R U_L < 0.44 W/m^2 \text{ } ^\circ C = 11\%$; natural convection heat exchanger system: summer, $F_R(\tau\alpha) < 0.013 = 2.3\%$, $F_R U_L < 1 W/m^2 \text{ } ^\circ C = 27\%$, winter, $F_R(\tau\alpha) < 0.01 = 1.8\%$, $F_R U_L < 0.41 W/m^2 \text{ } ^\circ C = 11\%$).

B.2 Determining the Errors in Heat Exchanger Analysis

The heat exchanger effectiveness for the drain-back system is defined as in eqn (B.6). Typical values for the needed temperatures are: $T_{HX,out} = 38^\circ\text{C}$, $T_{HX,in} = 16^\circ\text{C}$, and $T_{TnkMax} = 44^\circ\text{C}$. Let x equal the maximum tank temperature minus the heat exchanger inlet temperature, which is 28°C , and let y equal the heat exchanger outlet temperature minus the heat exchanger inlet temperature, 22°C . The effectiveness is then 0.78.

$$\epsilon = \frac{(\dot{m}C_p)_{drw}(T_{HX,out} - T_{HX,in})}{(\dot{m}C_p)_{drw}(T_{TnkMax} - T_{HX,in})} = \frac{y}{x} = \frac{22}{28} = 0.78 \quad (\text{B.6})$$

The error associated with the effectiveness equation is

$$E_\epsilon = \frac{\sqrt{2}}{x^2} E_{Temp} (x^2 - xy + y^2)^{1/2} = 0.005 \quad (\text{B.7})$$

Which is about 0.6%.

The inside convective coefficient is given as:

$$h = 0.023 \frac{k}{v} \left(\frac{4\rho P r \dot{V}}{\pi} \right)^{0.8} D^{-1.8} = B \dot{V}^{0.8} D^{-1.8} \approx 2650 \text{ W/m}^2\text{C} \quad (\text{B.8})$$

The error is thus,

$$E_h = \left[(E_v B (10.8) \dot{V}^{-0.2} D^{-1.8})^2 + (E_D B (-1.8) \dot{V}^{0.8} D^{-2.8})^2 \right]^{1/2} = 207, \text{ or } 8\% \quad (\text{B.9})$$

The overall loss coefficient is given as eqn (B.10).

$$U = \frac{(\dot{V} \rho C_p)(T_{HX,out} - T_{HX,in})}{A \left(T_{Tnk} - \left(\frac{T_{HX,out} - T_{HX,in}}{2} \right) \right)} = \frac{(\dot{V} \rho C_p) \left(\frac{x}{y} \right)}{A} \approx 800 \text{ W/m}^2\text{C} \quad (\text{B.10})$$

The error is given as eqn (B.11).

$$E_U = \left[\left(\frac{E_V}{\dot{V}} U \right)^2 + ((\dot{V} \rho C_p) E_T)^2 \left(\frac{1}{y} \right)^4 \left(\frac{3}{2} x^2 \right) + 2y^2 + \left(\frac{E_A}{A} U \right)^2 \right]^{1/2} \approx 20, \text{ or } 2.5\% \quad (\text{B.11})$$

The energy obtained from the heat exchanger is the amount of heat that has been transferred to the draw water and is given as:

$$Q_{coil} = (\dot{V} \rho C_p)_{drw} (T_{HX, out} - T_{HX, in}) = 18,024 \text{ W} \quad (\text{B.12})$$

The error may then be written as:

$$E_{coil} = Q_{coil} \left[\left(\frac{E_V}{\dot{V}} \right)^2 + \left(\frac{2E_T}{T_{HX, out} - T_{HX, in}} \right)^2 \right]^{1/2} = 164, \text{ or } 1\% \quad (\text{B.13})$$

The outside convective coefficient for the coil is calculated as eqn (B.14).

$$h_o = \left[\frac{1}{U} - \frac{1}{h} \right]^{-1} = 1146 \text{ W/m}^2 \text{ } ^\circ\text{C} \quad (\text{B.14})$$

The associated error is then calculated as eqn (B.15).

$$E_{h_o} = \left[\left(E_U U^{-2} \left(\frac{1}{U} - \frac{1}{h} \right)^{-2} \right)^2 + \left(E_h h^{-2} \left(\frac{1}{U} - \frac{1}{h} \right)^{-2} \right)^2 \right]^{1/2} = 56, \text{ or } 5\% \quad (\text{B.15})$$

B.3 Determining Error in the UA Loss Coefficient for the Tanks

The heat loss coefficients for the hot water storage tanks is calculated from experimental data. The error in the time is given as 7 seconds, the error in the temperature difference is 0.14°C , and the error in the mass is 0.2 kg. The auxiliary tank temperature is constant at 55°C , and the ambient temperature is 20°C . The equation used to determine the loss coefficient is given as eqn (B.16). The experimental value for the auxiliary tanks is about $2 \text{ W/}^\circ\text{C}$. The value is about $3 \text{ W/}^\circ\text{C}$ for the main solar storage tank of the drain-back

system, and 5 W/°C for the main solar storage tank of the natural convection heat exchanger system.

$$UA = \frac{MC_p}{t} \ln(T_{Tank} - T_a) = \frac{MC_p}{t} \ln(\Delta T) \approx 2 \text{ W/°C} \quad (\text{B.16})$$

The error associated with this calculation is given as eqn (B.17) and is 0.003 W/°C for the auxiliary tanks, or about 0.12%. The error is 0.003 W/°C for the drain-back system main tank, which is about 0.1%, and the error is 0.004 W/°C for the natural convection heat exchanger system, or 0.07%.

$$E_{UA} = UA \left[\left(\frac{E_M}{M} \right)^2 + \left(\frac{E_t}{t} \right)^2 + \left(\frac{E_{\Delta T}}{\Delta T \ln \Delta T} \right)^2 \right]^{1/2} = 0.003, \text{ or } 0.12\% \quad (\text{B.17})$$

B.4 Determining the Error in Parasitic Losses

The losses from the pump can be determined using Bernoulli's equation, the final form of the equation is shown in eqn (B.18). The rise in temperature across the pump can be considered negligible.

$$\dot{W} = \dot{m} \left(\frac{\Delta P}{\rho} + \Delta z g \right) \quad (\text{B.18})$$

The elevation head is about 4.5 m. The work for each system is approximately 5.77 W for the drain-back system, 3.93 W for the natural convection heat exchanger system, and 6.86 W for the ICS system. The error can be calculated as

$$E_W = \left[E_m^2 \left(\frac{\Delta P}{\rho} + \Delta z g \right)^2 + \left(\frac{E_{\Delta P} \dot{m}}{\rho} \right)^2 + (E_{\Delta z} \dot{m} g)^2 \right]^{1/2} \quad (\text{B.19})$$

The errors associated with each system are 6.83e-3 (0.12%) for the drain-back system, 2.83e-3 (0.07%) for the natural convection heat exchanger system, and 0.0128 (0.19%) for the ICS system.

APPENDIX C

Program Listings

PROGRAM INFINITY

```
C
C This program will read the notebook data files
C and strip out all of the "infinite" numbers
C
    REAL var(87)
    CHARACTER file1*12,Time*15
    CHARACTER Junk1*13,Junk2*49,Junk3*834,Junk4*667
    CHARACTER*80 Junk1,Junk2,Junk3,Junk4
    DATA var/87*0.0/

C
    PRINT *, 'Input the file name to read'
    READ *, file1
    OPEN (unit=11,file=file1,err=700,status='old')
    OPEN (unit=12,file='syst.prn',err=700,status='new')

C
    DO 10 i=-3,1401
        IF (i.eq.-3) THEN
            READ (11,*,END=1010) Junk1
            WRITE (12,21) Junk1
        ELSEIF (i.eq.-2) THEN
            READ (11,*,END=1010) Junk2
            WRITE (12,22) Junk2
        ELSEIF (i.eq.-1) THEN
            READ (11,*,END=1010) Junk3
            WRITE (12,23) Junk3
        ELSEIF (i.eq.0) THEN
            READ (11,*,END=1010) Junk4
            WRITE (12,24) Junk4
        ELSE
            READ (11,*,END=1010) (var(j), j=1,87), Time DO 30 j=1,87
            IF (var(j).gt.1000.0) THEN
                var(j)=0.0
            ELSEIF (var(j).lt.-1000.0) THEN
                var(j)=0.0
            ENDIF
30      CONTINUE
            WRITE (12,25) (var(j), j=1,87), Time
        ENDIF
    10 CONTINUE
    21 FORMAT(' ',A11,' ')
    22 FORMAT(' ',A47,' ')
    23 FORMAT(' ',A834,' ')
    24 FORMAT(' ',A667,' ')
    25 FORMAT(' ',A80,' ')
```

```

23 FORMAT (A834)
24 FORMAT (A667)
25 FORMAT (87 (F10.6,1X),A15)
    GOTO 1010
700 PRINT*, 'ERROR when opening a file'
1010 STOP
    END

```

PROGRAM REDUCTDRAW

```

C
C This program is used to analyze the data from the DHW project.
C Each file contains 6 hours worth of data for 83 notebook channels.
C
C Variable Types
    REAL var(13,8),sumvar(13,8),avevar(13,8,3),Resis,rltim1,rltim
    REAL var1(13,8,3),dt,cdrw,rltim2
    DATA var/104*0.0/,sumvar/104*0.0/,avevar/312*0.0/,var1/312*0.0/
    INTEGER i,j,ihr,ichk,k,ihr1,isum,jsum,icnt,jcnt,jt,iavetim,jtim
    INTEGER imin,isec,ihour,tcnt,tcnt1,jt2,itim
C Variables for extracting draw data
    INTEGER iflwASN,iflwNEG,iflwTH,iwrite,iunit1,iunit2 C
    CHARACTER*12 File1(0:99)
    CHARACTER Nul*1, Nu2*2,Junk*80,Time*15
    CHARACTER date1*13,date2*30,csyst*13
C Process for Opening and Reading all of the data files.
    PRINT*, 'Have you checked your file names? If they are'
    PRINT*, 'correct, enter 1, If they are not, enter 2'
    READ*, ichk
    IF (ichk.eq.2) THEN
        GOTO 1000
    ENDIF
C Open the output files
C Output files for averaged data
    OPEN(unit=16,file='heading.dat',err=700,status='new')
    OPEN(unit=17,file='nonsyst.dat',err=700,status='new')
    OPEN(unit=18,file='asn.dat',err=700,status='new')
    OPEN(unit=19,file='neg.dat',err=700,status='new')
    OPEN(unit=20,file='thermo.dat',err=700,status='new')
C Output files for the draws, raw data
    OPEN(unit=21,file='nsystdrw.dat',err=700,status='new')
    OPEN(unit=22,file='asndrw.dat',err=700,status='new')
    OPEN(unit=23,file='negdrw.dat',err=700,status='new')
    OPEN(unit=24,file='thmndrw.dat',err=700,status='new')
C Output files for calculated energy data, no draw (averaged)
    OPEN(unit=51,file='asnengy.dat',err=700,status='new')
    OPEN(unit=53,file='negengy.dat',err=700,status='new')
    OPEN(unit=56,file='threngy.dat',err=700,status='new')
C Output files for calculated energy data, with draws (raw)
    OPEN(unit=52,file='aengydw.dat',err=700,status='new')
    OPEN(unit=54,file='nengydw.dat',err=700,status='new')
    OPEN(unit=57,file='tengydw.dat',err=700,status='new')
C
C Read in the averaging time
C 5 minute averaging, the data acquisition interval is 15 seconds.
    PRINT*, 'Enter the # of minutes (integer) you wish to average over'
    READ*, jtim
    iavetim=INT(jtim*60.0/15.0)

```

```

C
C Input the number of sets of data files to be read.
C For six hours, there will be 1440 lines of data. There may be less
C if the file is not six full hours.
    PRINT*, 'Enter the number on the first set of data files'
    PRINT*, 'to run'
    READ*, ihr
    5 PRINT*, 'Enter the number on the last set of data files'
    PRINT*, 'to run'
    READ*, ihrl
C Putting the headings on the output files:
    WRITE(16,*) '#File Set, Date and Time heading'
    WRITE(16,*) ''
C
    DO 15 i=17,24
        IF ((i.eq.17).OR.(i.eq.21)) THEN
            WRITE(i,*)
+           'Non-system Specific data, Temps=deg C, Flows=m^3/s, Pyran=W/m^2'
            WRITE(i,*)
+           'WdSpd=m/s,WdDir=Deg(44E,134S,224W,314N),nb time,avetime,clktime'
            WRITE(i,*) ''
            WRITE(i,1)
        ENDIF
        1 FORMAT('IndrAmb',T9,',',SupplyT',T17,',',DrainT',T25,',',OutAmb',T33,
+       ',MxDwT',T41,',',SupFlw',T54,',',DrnFlw',T67,',',PyHor',T80,
+       ',PyTlt',T93,',',WdSpd',T106,',',WdDir',T119,',',Time') C
        IF ((i.eq.18).OR.(i.eq.22)) THEN
            WRITE(i,*) 'ASN System'
            WRITE(i,*)
+           'Temps=Deg C,Flows=m^3/s,Pressure=N/m^2, Power=Watts'
            WRITE(i,*) ''
            WRITE(i,2)
        ENDIF
        2 FORMAT('InAux',T9,',',InPmp',T17,',',BkPmp',T25,',',Bkcol',T33,
+       ',O/SBkCl',T41,',',O/SToCl',T49,',',MainBot',T57,',',Main7',T65,
+       ',Main6',T73,',',Main5',T81,',',Main4',T89,',',Main3',T97,',',Main2',
+       ',T105,',',MainTop',T113,',',AuxBot',T121,',',Aux7',T129,',',Aux6',T137,
+       ',Aux5',T145,',',Aux4',T153,',',Aux3',T161,',',Aux2',T169,',',AuxTop',
+       ',T177,',',Flow',T190,',',delP',T203,',',PmpX',T216,',',TnkX',T229,
+       ',DwFlwS',T242,',',Time')
C
        IF ((i.eq.19).OR.(i.eq.23)) THEN
            WRITE(i,*) 'NEG System'
            WRITE(i,*)
+           'Temps=Deg C,Flows=m^3/s,Pressure=N/m^2, Power=Watts'
            WRITE(i,*) ''
            WRITE(i,3)
        ENDIF
        3 FORMAT('InTnk',T9,',',BkTnk',T17,',',InPmp',T25,',',BkPmp',T33,
+       ',O/SToCl',T41,',',O/SBkCl',T49,',',TankBot',T57,',',Tank7',T65,
+       ',Tank6',T73,',',Tank5',T81,',',Tank4',T89,',',Tank3',T97,',',Tank2',
+       ',T105,',',TankTop',T113,',',Flow',T126,',',delP',T139,',',PmpX',T152,
+       ',TnkX',T165,',',DwFlwS',T178,',',Time')
C
        IF ((i.eq.20).OR.(i.eq.24)) THEN
            WRITE(i,*) 'Thermodynamics System'
            WRITE(i,*)
+           'Temps=Deg C,FTIFlw=m^3/s,MicFlw=kg/s, Pres=N/m^2, Pwr=Watts'
            WRITE(i,*) ''

```

```

        WRITE(i,4)
    ENDIF
4  FORMAT('ToCol',T9,',BkCol',T17,',InMTnk',T25,',BkMTnk',T33,
+',InATnk',T41,',O/SToCl',T49,',O/SBkCl',T57,',MainBot',T65,+',Main7',T73,
+',Main6',T81,',Main5',T89,',Main4',T97,',Main3',T105,',Main2',
+',T113,',MainTop',T121,',AuxBot',T129,',Aux7',T137,',Aux6',T145,
+',Aux5',T153,',Aux4',T161,',Aux3',T169,',Aux2',T177,',AuxTop',
+',T185,',FlwFTI',T198,',FlwMic',T211,',delP',T224,',PmpX',
+',T237,',TnkX',T250,',FlwDir',T263,',DwFlwS',T276,',Time')
15 CONTINUE
C Headings for the calculated energy files
    WRITE(51,*)'ASN System, Energy Calcs, no draws' WRITE(51,*)''
    WRITE(51,31)
31 FORMAT('SolTnk',T14,',SolCol',T28,',PmpPow',T42,',VlMTkT',T56,
+',VlATkT',T70,',MTkLoss',T84,',ATkLoss',T98,',MTk55',T112,
+',ATk55',T126,',BlkCapM',T140,',NdlCapM',T154,',BlkCapA',T168,
+',NdlCapA',T182,',Coll_dt',T196,',Coll_Eff',T210,',FrU1',T224,
+',FrTa',T238,',Time')
    WRITE(52,*)'ASN System, Energy Calcs, with draws' WRITE(52,*)''
    WRITE(52,32)
32 FORMAT('SolTnk',T14,',SolCol',T28,',PmpPow',T42,',VlMTkT',T56,
+',VlATkT',T70,',MTkLoss',T84,',ATkLoss',T98,',MTk55',T112,
+',ATk55',T126,',BlkCapM',T140,',NdlCapM',T154,',BlkCapA',T168,
+',NdlCapA',T182,',SolAux',T196,',DrwAux',T210,',SolDrw',T224,
+',Drw55',T238,',SSFThr',T252,',SSFTRN',T266,',MaxT',T280,',EffAve',
+',T294,',EffMax',T308,',EffTop',T322,',LMTDAve',T336,',LMTDMax',
+',T350,',LMTDTop',T364,',Coll_dt',T378,',Coll_Eff',T392,',FrU1',
+',T406,',FrTa',T420,',Time')
C
    WRITE(53,*)'NEG System, Energy Calcs, no draws' WRITE(53,*)''
    WRITE(53,33)
33 FORMAT('SolTnk',T14,',SolCol',T28,',PmpPow',T42,',VlATkT',T56,',AT
+kLoss',T70,',ATk55',T84,',BlkCapA',T98,',NdlCapA',T112,
+',Coll_dt',T126,',Coll_Eff',T140,',FrU1',T154,+',FrTa',T168,',Time')
    WRITE(54,*)'NEG System, Energy Calcs, with draws' WRITE(54,*)''
    WRITE(54,34)
34 FORMAT('SolTnk',T14,',SolCol',T28,',PmpPow',T42,',VlATkT',T56,
+',ATkLoss',T70,',ATk55',T84,',BlkCapA',T98,',NdlCapA',T112,
+',DrwCol',T126,',DrwAux',T140,',SolDrw',T154,',Drw55',T168,
+',DrwTnk',T182,',SSFThr',T196,',SSFTRN',T212,',
+',Coll_dt',T226,',Coll_Eff',T240,',FrU1',T256,+',FrTa',T270,',Time')
C
    WRITE(56,*)'Thermo Dynamics System, Energy Calcs, no draws' WRITE(56,*)''
    WRITE(56,36)
36 FORMAT('SolTnk',T14,',SolCol',T28,',PmpPow',T42,',SolGly',T56,
+',VlMTkT',T70,',VlATkT',T84,',MTkLoss',T98,',ATkLoss',T112,
+',MTk55',T126,',ATk55',T140,',BlkCapM',T154,',NdlCapM',T168,',BlkC
+apA',T182,',NdlCapA',T196,',EffGly',T210,',EffWat',T224,
+',Coll_dt',T238,',Coll_Eff',T252,',FrU1',T266,+',FrTa',T280,',Time')
    WRITE(57,*)'Thermo Dynamics System, Energy Calcs, with draws' WRITE(57,*)''
    WRITE(57,37)
37 FORMAT('SolTnk',T14,',SolCol',T28,',PmpPow',T42,',SolGly',T56,
+',VlMTkT',T70,',VlATkT',T84,',MTkLoss',T98,',ATkLoss',T112,
+',MTk55',T126,',ATk55',T140,',BlkCapM',T154,',NdlCapM',T168,
+',BlkCapA',T182,',NdlCapA',T196,',EffGly',T210,',EffWat',T224,
+',SolAux',T238,',DrwAux',T252,',SolDrw',T266,',Drw55',T280,
+',SSFThr',T294,',SSFTRN',T336,
+',Coll_dt',T322,',Coll_Eff',T336,',FrU1',T350,+',FrTa',T364,',Time')
C These are counters used to perform the averaging.

```



```

        icnt=0
        jcnt=0
C counters for extracting draw data
        iflwASN=0
        iflwNEG=0
        iflwTH=0
        iwrite=0
        tcnt=0
        tcnt1=0
        itim=0
C Loop for each "Notebook" file
        DO 10 i=ihr,ihr1
C
C This section opens all of the input files
        IF (i.le.9) THEN
            WRITE (Nu1, '(I1)') i
            File1(i)='norm'//nu1//'.prn'
        ELSE
            WRITE (nu2, '(I2)') i
            File1(i)='norm'//nu2//'.prn'
        ENDIF
        OPEN (unit=11, file=file1(i), err=700, status='old')
C This loop reads in the necessary information, one data
C line from each file at a time
        DO 20 j=-3,2880
C
C Reads in the heading for date and time. These will
C be printed to Heading.prn output file for reference.
        IF (j.le.0) THEN
            IF (j.eq.-2) THEN
                READ (11,22) date1, ihour, imin, isec, date2
                WRITE (16,55) i, date1, ihour, imin, isec, date2 ELSE
                READ (11,21) Junk
            ENDIF
        ELSE
C Determining clock time
            itim=itim+1
            rltim1=REAL(ihour)+REAL(imin)/60.0+REAL(isec)/3600.0
            IF (i.eq.ihr) THEN
                rltim2=REAL(ihour)+REAL(imin)/60.0+REAL(isec)/3600.0
            ENDIF
            rltim=rltim2+itim*15.0/3600.0
C
C Reading the data lines:
            READ (11,23,END=30,ERR=700) var(3,1), var(1,1), var(1,2),
+               var(1,3), var(10,1), var(10,2), var(10,3), var(10,4), var(10,5),
+               var(7,1), var(7,2), var(7,3), var(7,4), var(3,2), var(3,3),
+               var(3,4), var(1,4), var(10,6), var(10,7),
+               var(7,5), var(7,6), var(3,5), var(3,6), var(8,1), var(8,2),
+               var(8,3), var(8,4), var(8,5), var(8,6), var(8,7), var(8,8), var(1,5),
+               var(11,1), var(11,2), var(11,3),
+               var(11,4), var(11,5), var(11,6), var(11,7), var(11,8), var(12,1),
+               var(12,2), var(12,3), var(12,4), var(12,5), var(12,6), var(12,7),
+               var(12,8), var(4,1), var(4,2), var(4,3),
+               var(4,4), var(4,5), var(4,6), var(4,7), var(4,8), var(5,1),
+               var(5,2), var(5,3), var(5,4), var(5,5), var(5,6), var(5,7),
+               var(5,8), var(13,1), var(13,2), var(9,1),
+               var(6,1), var(2,1), var(2,2), var(13,3), var(9,2), var(6,2),
+               var(13,4), var(9,3), var(6,3), var(13,5), var(9,4), var(6,4),

```

```

+          var(13,6),var(2,3),var(2,4),var(2,5),var(2,6),var(6,5),
+          var(9,5),var(13,7),Time
C
C This section calls the subroutines that are used to convert the
C voltage readings into the appropriate output (SI units).
      CALL CNVRT2(var)
      CALL CNVRT6(var)
      CALL CNVRT9(var)
      CALL CNVRT13(var,i,ihr,Resis,j)
      icnt=icnt+1
C This section strips out the data from the draw and puts them
C in appropriate files; no averaging.
C Sets all required variables:NEG
      IF(var(9,5).gt.10.0)THEN
        iflwNEG=iflwNEG+1
        csyst='NEG System'
        iwrite=23
        IF(iflwNEG.eq.1)THEN
          cdrw=rltim
        ENDIF
C :Thermo
      ELSEIF(var(13,7).gt.10.0)THEN
        iflwTH=iflwTH+1
        csyst='Thermo System'
        iwrite=24
C :ASN
      ELSEIF(var(6,5).gt.10.0)THEN
        iflwASN=iflwASN+1
        csyst='ASN System'
        iwrite=22
      ENDIF
C Writes out file headings
      IF((iflwNEG.eq.1).OR.(iflwTH.eq.1).OR.(iflwASN.eq.1))THEN
        WRITE(21,*)''
        WRITE(21,*)Date2,file1(i),j
        WRITE(21,*)'The time is ',cdrw
        WRITE(21,*)'',csyst
        WRITE(iwrite,*)''
        WRITE(iwrite,*)Date2,file1(i),j
        WRITE(iwrite,*)'The time is ',cdrw
      ENDIF
C Writes data to appropriate files
C Non-system specific data
      IF((iflwNEG.gt.0).OR.(iflwTH.gt.0).OR.(iflwASN.gt.0))THEN
C Time counter set up for Energy Subroutine
        tcnt=tcnt+1
        IF(tcnt.gt.3)THEN
          tcnt=1
        ENDIF
C Loops to set draw variables to read in variables
        DO 41 jt=1,13
          DO 42 jt2=1,8
            var1(jt,jt2,tcnt)=var(jt,jt2)
          42 CONTINUE
        41 CONTINUE
C
        WRITE(21,51)(var1(1,jt,tcnt),jt=1,5),
+          (var1(2,jt,tcnt),jt=1,6),Time,icnt*8,rltim
C NEG system data

```

```

        IF (iflwNEG.gt.0) THEN
            WRITE (23,53) (var1(7,jt,tcnt),jt=1,6),
+               (var1(8,jt,tcnt),jt=1,8),
+               (var1(9,jt,tcnt),jt=1,5),rltim
C Calls the NEG energy subroutine and sets delta time
            dt=15.0
            iunit1=53
            iunit2=54
            CALL ENERGYN(var1,tcnt,dt,iunit1,iunit2,rltim,iflwNEG)
C Thermo Dynamics system data
            ELSEIF (iflwTH.gt.0) THEN
                WRITE (24,54) (var1(10,jt,tcnt),jt=1,7),
+               (var1(11,jt,tcnt),jt=1,8),
+               (var1(12,jt,tcnt),jt=1,8),
+               (var1(13,jt,tcnt),jt=1,7),rltim
C Calls the Thermo energy subroutine and sets delta time
                dt=15.0
                iunit1=56
                iunit2=57
                CALL ENERGYT(var1,tcnt,dt,iunit1,iunit2,rltim,iflwTH)
C ASN system data
                ELSEIF (iflwASN.gt.0) THEN
                    WRITE (22,52) (var1(3,jt,tcnt),jt=1,6),
+               (var1(4,jt,tcnt),jt=1,8),
+               (var1(5,jt,tcnt),jt=1,8),
+               (var1(6,jt,tcnt),jt=1,5),rltim
C Calls the ASN energy subroutine and sets delta time
                    dt=15.0
                    iunit1=51
                    iunit2=52
                    CALL ENERGYA(var1,tcnt,dt,iunit1,iunit2,rltim,iflwASN)
                ENDIF
            ENDIF
C Resets necessary variables
            IF ((var(9,5).lt.10.0).AND.(iflwNEG.gt.0)) THEN
                iflwNEG=0
                tcnt=0
            ELSEIF ((var(13,7).lt.10.0).AND.(iflwTH.gt.0)) THEN
                iflwTH=0
                tcnt=0
            ELSEIF ((var(6,5).lt.10.0).AND.(iflwASN.gt.0)) THEN
                iflwASN=0
                tcnt=0
            ENDIF
C
C These loops perform the summing operation for all the variables
C The absolute value function is used on the voltage readings to
C ensure they were positive
C
            DO 70 isum=1,13
                DO 80 jsum=1,8
                    sumvar(isum,jsum)=sumvar(isum,jsum)+var(isum,jsum)
                80 CONTINUE
            70 CONTINUE
C
C The averaging is accomplished
            IF (icnt.eq.iavetim.or.jcnt.eq.icnt) THEN
C Time counter set up for Energy Subroutine
                tcnt1=tcnt1+1

```

```

        IF (tcnt1.gt.3) THEN
            tcnt1=1
        ENDIF

C
        DO 90 isum=1,13
            DO 100 jsum=1,8
                avevar(isum,jsum,tcnt1)=sumvar(isum,jsum)/REAL(icnt)
            100    CONTINUE
        90    CONTINUE

C
C Writing the averaged values to the output files. The averaging
C time interval is also written.
C
        WRITE(17,51) (avevar(1,jt,tcnt1),jt=1,5),
+           (avevar(2,jt,tcnt1),jt=1,6),Time,icnt*8,rltim
        WRITE(18,52) (avevar(3,jt,tcnt1),jt=1,6),
+           (avevar(4,jt,tcnt1),jt=1,8),
+           (avevar(5,jt,tcnt1),jt=1,8),
+           (avevar(6,jt,tcnt1),jt=1,5),rltim
        WRITE(19,53) (avevar(7,jt,tcnt1),jt=1,6),
+           (avevar(8,jt,tcnt1),jt=1,8),
+           (avevar(9,jt,tcnt1),jt=1,5),rltim
        WRITE(20,54) (avevar(10,jt,tcnt1),jt=1,7),
+           (avevar(11,jt,tcnt1),jt=1,8),
+           (avevar(12,jt,tcnt1),jt=1,8),
+           (avevar(13,jt,tcnt1),jt=1,7),rltim
C Calls the energy subroutines sequentially (ASN, NEG, Thermo)
C and sets the delta time to 5 minutes
        dt=5.0*60.0
        iunit1=51
        iunit2=52
        CALL ENERGYA(avevar,tcnt1,dt,iunit1,iunit2,rltim,iflwASN)
        iunit1=53
        iunit2=54
        CALL ENERGYN(avevar,tcnt1,dt,iunit1,iunit2,rltim,iflwNEG)
        iunit1=56
        iunit2=57
        CALL ENERGYT(avevar,tcnt1,dt,iunit1,iunit2,rltim,iflwTH)

C
C Reinitializes added variables to zero once the averaging
C has been accomplished
        icnt=0
        DO 40 isum=1,13
            DO 50 jsum=1,8
                sumvar(isum,jsum)=0.0
            50    CONTINUE
        40    CONTINUE
        ENDIF
    20    CONTINUE
    20    CONTINUE

C
C Line 30 is for when the end of file has been reached. If it is
C the last data file, then averaging and writing of those last
C data points needs to be accomplished, otherwise, just go to "10
C continue" and read the next data file.
    30    IF((i.eq.ihrl).AND.(icnt.lt.iavetim)) THEN
C Time counter set up for Energy Subroutine
        tcnt1=tcnt1+1
        IF (tcnt1.gt.3) THEN

```

```

        tcnt1=1
    ENDIF
C
        DO 110 isum=1,13
            DO 120 jsum=1,8
                DO 130 jt=1,3
                    avevar(isum,jsum,jt)=sumvar(isum,jsum)/REAL(icnt)
130                CONTINUE
120            CONTINUE
110        CONTINUE
C Writing the averaged values to the output files. The averaging
C time interval is also written.
C
        WRITE(17,51) (avevar(1,jt,tcnt1),jt=1,5),
+            (avevar(2,jt,tcnt1),jt=1,6),Time,icnt*8,rltim
        WRITE(18,52) (avevar(3,jt,tcnt1),jt=1,6),
+            (avevar(4,jt,tcnt1),jt=1,8),
+            (avevar(5,jt,tcnt1),jt=1,8),
+            (avevar(6,jt,tcnt1),jt=1,5),rltim
        WRITE(19,53) (avevar(7,jt,tcnt1),jt=1,6),
+            (avevar(8,jt,tcnt1),jt=1,8),
+            (avevar(9,jt,tcnt1),jt=1,5),rltim
        WRITE(20,54) (avevar(10,jt,tcnt1),jt=1,7),
+            (avevar(11,jt,tcnt1),jt=1,8),
+            (avevar(12,jt,tcnt1),jt=1,8),
+            (avevar(13,jt,tcnt1),jt=1,7),rltim
C Calls the energy subroutines sequentially (ASN, NEG, Thermo)
C and sets the delta time to 5 minutes
        dt=5.0*60.0
        iunit1=51
        iunit2=52
        CALL ENERGYA(avevar,tcnt1,dt,iunit1,iunit2,rltim,iflwASN)
        iunit1=53
        iunit2=54
        CALL ENERGYN(avevar,tcnt1,dt,iunit1,iunit2,rltim,iflwNEG)
        iunit1=56
        iunit2=57
        CALL ENERGYT(avevar,tcnt1,dt,iunit1,iunit2,rltim,iflwTH)
C
        ENDIF
    10 CONTINUE
C
C Format Statements
21 FORMAT(A80)
22 FORMAT(A13,I2,1X,I2,1X,I2,A30)
23 FORMAT(87(F10.6,1X),A15)
51 FORMAT(5(F7.3,1X,','),6(E12.6,1X,','),A15,2X,',',I5,2X,
+    ', ',F7.3)
52 FORMAT(22(F7.3,1X,','),5(E12.6,1X,','),F7.3)
53 FORMAT(14(F7.3,1X,','),5(E12.6,1X,','),F7.3)
54 FORMAT(23(F7.3,1X,','),7(E12.6,1X,','),F7.3)
55 FORMAT(I3,2X,A13,I2,':',I2,':',I2,A30)
    GOTO 1010
700 PRINT*,'Error occurred reading file i=',i,'line j=',j
1000 PRINT*,'Please fix your files and try again' 1010 STOP
    END
C
C

```

SUBROUTINE CNVRT2(var2)

C This subroutine does the conversions of the supply and drain
C flows voltages, and the weather station voltages to the C required SI units used in
calculations. C

REAL var2(13,8)

C Converts voltages to zero if they are too small

DO 10 i=3,4

IF (var2(2,i).lt.0.0) THEN

var2(2,i)=0.0

ENDIF

10 CONTINUE

C Calculations/Conversions

var2(2,1)=(var2(2,1)+0.355)/1.173

var2(2,1)=var2(2,1)*0.0037854/60.0

var2(2,2)=(var2(2,2)-0.025)/0.070

var2(2,2)=var2(2,2)*0.0037854/60.0

var2(2,3)=(var2(2,3)+0.00004)/1.063813E-05

var2(2,4)=(var2(2,4)+0.00003)/9.963726E-06

var2(2,5)=0.89448+0.053519*(1.028*(var2(2,5))**3+
+ 1.506*(var2(2,5))**2+10.803*var2(2,5)+10.648)

var2(2,6)=-1.117*var2(2,6)**2+64.735*var2(2,6)+0.414

C This converts the flows to zero if they are negative

IF (var2(2,1).le.1.0E-04) THEN

var2(2,1)=0.0

ENDIF

IF (var2(2,2).le.1.5E-05) THEN

var2(2,2)=0.0

ENDIF

DO 20 i=3,4

IF (var2(2,i).lt.5.0) THEN

var2(2,i)=0.0

ENDIF

20 CONTINUE

RETURN

END

C

C

SUBROUTINE CNVRT6(var2)

C This subroutine converts voltages from ASN flow meters,

C Pressure transducers, and Watt transducers to SI units.

REAL var2(13,8)

C Converts voltages to zero if they are too small

IF (var2(6,2).le.0.0008) THEN

var2(6,2)=0.0

ENDIF

IF (var2(6,3).le.0.0002) THEN

var2(6,3)=0.0

ENDIF

IF (ABS(var2(6,4)).le.0.00003) THEN

var2(6,4)=0.0

ENDIF

C Calculations/Conversions

var2(6,1)=(var2(6,1)-0.503)/2.068

var2(6,1)=var2(6,1)*0.0037854/60.0

var2(6,2)=(15.0/0.1)*var2(6,2)

var2(6,2)=var2(6,2)*6894.8

var2(6,3)=(20000.0/1.154)*var2(6,3)

var2(6,4)=(400000.0)*ABS(var2(6,4))

C This converts the flows to zero if they are negative

```

        IF (var2(6,1).le.1.75E-05) THEN
            var2(6,1)=0.0
        ENDIF
        RETURN
    END

C
C
    SUBROUTINE CNVRT9(var2)
C This subroutine converts voltages from NEG flow meters,
C Pressure transducers, and Watt transducers to SI units.
    REAL var2(13,8)
C Converts voltages to zero if they are too small
    IF (var2(9,2).le.0.0029) THEN
        var2(9,2)=0.0
    ENDIF
    IF (var2(9,3).le.0.00015) THEN
        var2(9,3)=0.0
    ENDIF
    IF (ABS(var2(9,4)).le.0.0050) THEN
        var2(9,4)=0.0
    ENDIF
C Calculations/Conversions
    var2(9,1)=(var2(9,1)-0.053)/0.063
    var2(9,1)=var2(9,1)*0.0037854/60.0
    var2(9,2)=(15.0/0.1)*var2(9,2)
    var2(9,2)=var2(9,2)*6894.8
    var2(9,3)=(20000.0/1.177)*var2(9,3)
    var2(9,4)=(400000.0/200.355)*ABS(var2(9,4))
C This converts the flows to zero if they are negative
    IF (var2(9,1).le.5.0E-06) THEN
        var2(9,1)=0.0
    ENDIF
    RETURN
    END

C
C
    SUBROUTINE CNVRT13(var2,i,ihr,Resis,j)
C This subroutine converts voltages from Thermodynamics flow meters,
C Pressure transducers, and Watt transducers to SI units.
    REAL var2(13,8),Resis
    INTEGER i,ihr,j
C
C Read in the Resistance of the Micromotion Flowmeter'
    IF ((i.eq.ihr).AND.(j.le.1)) THEN
        PRINT*, 'Enter the Micromotion Flowmeter resistance'
        PRINT*, '(ohms), for your data files if you want to use'
        PRINT*, 'the default (477), enter 1.0 (one). This value'
        PRINT*, 'is used for all the file sets.'
        READ*,Resis
        IF (Resis.eq.1.0) THEN
            Resis=477.0
        ENDIF
    ELSE
        Resis=Resis
    ENDIF
C Converts voltages to zero if they are too small
    IF (var2(13,1).ge.3.0) THEN
        var2(13,1)=0.0
    ENDIF

```

```

      IF (var2(13,3) .le. 0.0) THEN
        var2(13,3)=0.0
      ENDIF
      IF (var2(13,4) .le. 0.0) THEN
        var2(13,4)=0.0
      ENDIF
      IF (ABS (var2(13,5)) .le. 0.00015) THEN
        var2(13,5)=0.0
      ENDIF
C Calculations/Conversions
      var2(13,1)=(var2(13,1)+0.013)/3.862
      var2(13,1)=var2(13,1)*0.0037854/60.0
C Micro Motion Flow meter is in lb/min, not gal/min!
      var2(13,2)=(var2(13,2)/Resis)-0.0041)*(10.0/0.016)
var2(13,2)=var2(13,2)*(0.45359/60.0)
      var2(13,3)=(30.0/0.1)*var2(13,3)
      var2(13,3)=var2(13,3)*6894.8
      var2(13,4)=(20000.0/0.65)*(var2(13,4)+0.0002)
      var2(13,5)=(400000.0/2.101)*ABS(var2(13,5))
C This converts the flows to zero if they are negative
      IF (var2(13,1) .le. 3.0E-06) THEN
        var2(13,1)=0.0
      ENDIF
      IF (var2(13,2) .le. 0.0) THEN
        var2(13,2)=0.0
      ENDIF
      IF (var2(13,4) .le. 6.5) THEN
        var2(13,4)=0.0
      ENDIF
      RETURN
    END

C
C
C Subroutine to do energy calculations for ASN system
      SUBROUTINE ENERGY (var3,tcnt,dt,iunit1,iunit2,rltim,iflwASN)
C
      REAL var3(13,8,3),dt,rltim
      REAL ttop(2,3),t7(2,3),t6(2,3),t5(2,3)
      REAL t4(2,3),t3(2,3),t2(2,3),tbot(2,3)
      REAL vlmkt(3),vlatkt(3),pytilt(3),outamb(3)
      REAL pmpflw(3),intnk,bktnk,incol(3),bkcol(3),delp,inaux
      REAL volmbot,volmtop,uam,uaa,volabot,volatop
      REAL volumem,volumea,inamb,drwflw,drwt,supt,t
      REAL soltnk,pmpow,mtkloss,atkloss,mtk55
      REAL atk55,blkcapm,ndlcapm,blkcapa,ndlcapa,solaux
      REAL drwaux,soldrw,drw55,ssfthr,ssftzn,maxt,effave
      REAL effmax,efftop,lmtdave,lmtdmax,lmtdtop
      REAL acnet,qu(3),coleff(3),coldt(3),frul,frta
      INTEGER it,itpl,itml,tcnt,i,j,k,kl,k2,k3,iunit1,iunit2,iflwASN
C Sets t-1, t, and t+1 from main program
      it=tcnt
      itpl=tcnt+1
      itml=tcnt-1
      IF (itpl.eq.4) THEN
        itpl=1
      ENDIF
      IF (itml.eq.0) THEN
        itml=3
      ENDIF

```



```

C Sets the system variables
C ASN System
    intnk=var3(3,4,it)
    bktnk=var3(3,2,it)
    delp=var3(6,2,it)
    inaux=var3(3,1,it)
    volmbot=0.04503
    volmtop=0.031
    i=4
    j=5
    uam=2.93
    uaa=2.04
    acnet=5.52
C The next two do loops initialize the tank
C temperatures and the variables used to calculate
C the collector efficiency: at (k1=1) t-1, (k1=2)t, (k1=3)t+1.
C The outside loop determines (k3=i) Main tank C and (k3=j) Auxiliary tank.
    DO 10 k1=1,3
        IF (k1.eq.1) THEN
            k2=itp1
        ELSEIF (k1.eq.2) THEN
            k2=itml
        ELSE
            k2=it
        ENDIF
        pmpflw(k1)=var3(6,1,k2)
        pytilt(k1)=var3(2,4,k2)
        outamb(k1)=var3(1,4,k2)
        incol(k1)=var3(3,6,k2)
        bkcol(k1)=var3(3,5,k2)
        DO 20 k=1,2
            IF (k.eq.1) THEN
                k3=i
            ELSE
                k3=j
            ENDIF
            ttop(k,k1)=var3(k3,8,k2)
            t7(k,k1)=var3(k3,7,k2)
            t6(k,k1)=var3(k3,6,k2)
            t5(k,k1)=var3(k3,5,k2)
            t4(k,k1)=var3(k3,4,k2)
            t3(k,k1)=var3(k3,3,k2)
            t2(k,k1)=var3(k3,2,k2)
            tbot(k,k1)=var3(k3,1,k2)
        20    CONTINUE
    10    CONTINUE
C Non-system specific variables
    volabot=0.01858
    volatop=0.009292
    volumem=volmbot*7+volmtop
    volumea=volabot*7+volatop
    inamb=var3(1,1,it)
    drwflw=var3(2,2,it)
    drwt=var3(1,3,it)
    supt=var3(1,2,it)
C Starting Calculations
C Solar into Storage Tank
    t=(intnk+bktnk)/2.0
    CALL RhocpW(t,cp,den)

```

```

      soltnk=pmpflw(3)*cp*den*(intnk-bktnk)
C Pumping Power
      pmpow=pmpflw(3)*delp
C Volume Average Tank Temperatures & collector
C Efficiency stuff at t-1, t, t+1
      DO 30 k=1,3
          vlmktk(k)=(tbot(1,k)+t2(1,k)+t3(1,k)+t4(1,k)+
+               t5(1,k)+t6(1,k)+t7(1,k))*volmbot+
+               ttop(1,k)*volmtop)/volumem
          vlatkt(k)=(tbot(2,k)+t2(2,k)+t3(2,k)+t4(2,k)+
+               t5(2,k)+t6(2,k)+t7(2,k))*volabot+
+               ttop(2,k)*volatop)/volumea
C Solar across Collectors
          t=(incol(k)+bkcol(k))/2.0
          CALL RhocpW(t,cp,den)
          qu(k)=pmpflw(k)*cp*den*(bkcol(k)-incol(k))
          coleff(k)=qu(k)/(acnet*pytilt(k))
          coldt(k)=(incol(k)-outamb(k))/pytilt(k)
30 CONTINUE
C Collector Efficiency slope (FrUl) and intercept (FrTa)
      frul=-1*((coleff(3)-coleff(2))/(coldt(3)-coldt(2)))
      frta=coleff(3)+frul*coldt(3)
C Tank Losses at time t
      mtkloss=uam*(vlmktk(3)-inamb)
      atkloss=uaa*(vlatkt(3)-inamb)
      mtk55=uam*(55-inamb)
      atk55=uaa*(55-inamb)
C Tank Capacitance a) by Bulk Volume,
C b) by nodes (MCp dT/dt, integration)
      CALL RhocpW(vlmktk(2),cp,den)
      blkcapm=volumem*cp*den*(vlmktk(3)-vlmktk(1))/(2.0*dt)
      ndlcapm=cp*den*((tbot(1,3)-tbot(1,1))+(t2(1,3)-t2(1,1))+
+ (t3(1,3)-t3(1,1))+(t4(1,3)-t4(1,1))+(t5(1,3)-t5(1,1))+
+ (t6(1,3)-t6(1,1))+(t7(1,3)-t7(1,1))*volmbot+(ttop(1,3)-
+ ttop(1,1))*volmtop)/(2.0*dt)
      CALL RhocpW(vlatkt(2),cp,den)
      blkcapa=volumea*cp*den*(vlatkt(3)-vlatkt(1))/(2.0*dt)
      ndlcapa=cp*den*((tbot(2,3)-tbot(2,1))+(t2(2,3)-t2(2,1))+
+ (t3(2,3)-t3(2,1))+(t4(2,3)-t4(2,1))+(t5(2,3)-t5(2,1))+
+ (t6(2,3)-t6(2,1))+(t7(2,3)-t7(2,1))*volabot+(ttop(2,3)-
+ ttop(2,1))*volatop)/(2.0*dt)
C
C Calculations to be performed during draws only
      IF(dt.eq.15.0) THEN
C Draw energy into auxiliary tank
          t=(inaux+supt)/2.0
          CALL RhocpW(t,cp,den)
          solaux=drwflw*den*cp*(inaux-supt)
C Draw energy out of auxiliary tank
          t=(drwt+inaux)/2.0
          CALL RhocpW(t,cp,den)
          drwaux=drwflw*den*cp*(drwt-inaux)
C Total draw energy
          t=(drwt+supt)/2.0
          CALL RhocpW(t,cp,den)
          soldrw=drwflw*den*cp*(drwt-supt)
C Additional Draw energy if tank were kept at 55C
          t=(55+drwt)/2.0
          CALL RhocpW(t,cp,den)

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drw55=drwflw*den*cp*(55-drwt)
C Solar Savings Fraction - Thermal
  ssfthr=soldrw/(soldrw+drw55+atk55-atkloss)
C Solar Savings Fraction - TRNSYS
  ssftrn=solaux/(solaux+drw55+atk55-atkloss)
C Find the Maximum main tank temperature
  maxT=0.0
  IF (ttop(1,3).gt.maxT) maxT=ttop(1,3)
  IF (t7(1,3).gt.maxT) maxT=t7(1,3)
  IF (t6(1,3).gt.maxT) maxT=t6(1,3)
  IF (t5(1,3).gt.maxT) maxT=t5(1,3)
  IF (t4(1,3).gt.maxT) maxT=t4(1,3)
  IF (t3(1,3).gt.maxT) maxT=t3(1,3)
  IF (t2(1,3).gt.maxT) maxT=t2(1,3)
  IF (tbot(1,3).gt.maxT) maxT=tbot(1,3)
C Heat Exchanger Calculations
C Effectiveness using average tank temperature
  effave=(inaux-supt)/(vlmtkt(3)-supt)
C Effectiveness using maximum tank temperature
  effmax=(inaux-supt)/(maxT-supt)
C Effectiveness using top tank temp.
  efftop=(inaux-supt)/(ttop(1,3)-supt)
C LMTD using average tank temperature
  lmtdave=((tbot(1,3)-supt)-(vlmtkt(3)-inaux))/(log((
+   tbot(1,3)-supt)/(vlmtkt(3)-inaux)))
C LMTD using maximum tank temperature
  lmtdmax=((tbot(1,3)-supt)-(maxT-inaux))/(log((
+   tbot(1,3)-supt)/(maxT-inaux)))
C LMTD using top tank temperature
  lmtdtop=((tbot(1,3)-supt)-(ttop(1,3)-inaux))/(log((
+   tbot(1,3)-supt)/(ttop(1,3)-inaux)))
  ENDIF
C Write the output to the appropriate files
  IF (dt.eq.15.0) THEN
    IF (iflwASN.eq.1) THEN
      WRITE(iunit2,*) ''
    ENDIF
    WRITE(iunit2,42) soltnk,qu(3),pmppow,vlmtkt(3),vlatkt(3),mtkloss,
+   atkloss,mtk55,atk55,blkcapm,ndlcapm,blkcapa,ndlcapa,solaux,
+   drwaux,soldrw,drw55,ssfthr,ssftrn,maxt,effave,effmax,
+   efftop,lmtdave,lmtdmax,lmtdtop,coldt(3),coleff(3),
+   frul,frta,rltim
  ELSE
    WRITE(iunit1,41) soltnk,qu(3),pmppow,vlmtkt(3),vlatkt(3),mtkloss,
+   atkloss,mtk55,atk55,blkcapm,ndlcapm,blkcapa,ndlcapa,
+   coldt(3),coleff(3),frul,frta,rltim
  ENDIF
41 FORMAT(17(E13.6,1X,', '),F7.3)
42 FORMAT(30(E13.6,1X,', '),F7.3)
  RETURN
END

C
C
C Subroutine to do energy calculations for NEG system
  SUBROUTINE ENERGYN(var3,tcont,dt,iunit1,iunit2,rltim,iflwNEG)
C
  REAL var3(13,8,3),dt,rltim
  REAL ttop(3),t7(3),t6(3),t5(3)
  REAL t4(3),t3(3),t2(3),tbot(3),coldt(3)

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REAL vlatkt(3),pytilt(3),outamb(3),qu(3),coleff(3)
REAL pmpflw(3),intnk,bktnk,incol(3),bkcol(3),delp,inaux
REAL uaa,volabot,volatop,frul,frta,acnet
REAL volumea,inamb,drwflw,drwt,supt,t
REAL soltnk,pmpow,atkloss
REAL atk55,blkcapa,ndlcapa,drwcol,drwaux
REAL soldrw,drw55,drwtmk,ssfthr,ssftrn
INTEGER it, itpl, itml, tcnt, j, k, k1, k2, iunit1, iunit2, iflwNEG
C Sets t-1, t, and t+1 from main program
it=tcnt
itpl=tcnt+1
itml=tcnt-1
IF (itpl.eq.4) THEN
    itpl=1
ENDIF
IF (itml.eq.0) THEN
    itml=3
ENDIF
C Sets the system variables
C NEG System
intnk=var3(7,1,it)
bktnk=var3(7,3,it)
delp=var3(9,2,it)
inaux=var3(7,1,it)
j=8
uaa=2.59
acnet=1.82
C The next two do loops initialize the tank
C temperatures and the collector efficiency
C variables at (k1=1)t-1, (k1=2)t, (k1=3)t+1.
DO 10 k1=1,3
    IF (k1.eq.1) THEN
        k2=itpl
    ELSEIF (k1.eq.2) THEN
        k2=itml
    ELSE
        k2=it
    ENDIF
    pmpflw(k1)=var3(9,1,k2)
    pytilt(k1)=var3(2,4,k2)
    outamb(k1)=var3(1,4,k2)
    incol(k1)=var3(7,5,k2)
    bkcol(k1)=var3(7,6,k2)
    ttop(k1)=var3(8,8,k2)
    t7(k1)=var3(8,7,k2)
    t6(k1)=var3(8,6,k2)
    t5(k1)=var3(8,5,k2)
    t4(k1)=var3(8,4,k2)
    t3(k1)=var3(8,3,k2)
    t2(k1)=var3(8,2,k2)
    tbot(k1)=var3(8,1,k2)
10 CONTINUE
C Non-system specific variables
volabot=0.01858
volatop=0.009292
volumea=volabot*7+volatop
inamb=var3(1,1,it)
drwflw=var3(2,2,it)
drwt=var3(1,3,it)

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      supt=var3(1,2,it)
C Starting Calculations
C Solar into Storage Tank
      t=(intnk+bktnk)/2.0
      CALL RhocpW(t,cp,den)
      soltnk=pmpflw(3)*cp*den*(intnk-bktnk)
C Pumping Power
      pmpow=pmpflw(3)*delp
C Volume Average Tank Temperatures & collector C efficiency stuff at t-1, t, t+1
      DO 30 k=1,3
          vlatkt(k)=(tbot(k)+t2(k)+t3(k)+t4(k)+t5(k)+
+           t6(k)+t7(k))*volabot+ttop(k)*volatop)/volumea
C Solar across Collectors
          t=(incol(k)+bkcol(k))/2.0
          CALL RhocpW(t,cp,den)
          qu(k)=pmpflw(k)*cp*den*(bkcol(k)-incol(k))
          coleff(k)=qu(k)/(acnet*pytilt(k))
          coldt(k)=(incol(k)-outamb(k))/pytilt(k)
      30 CONTINUE
C Collector Efficiency slope (FrU1) and intercept (FrTa)
      frul=-1*((coleff(3)-coleff(2))/(coldt(3)-coldt(2)))
      frta=coleff(3)+frul*coldt(3)
C Tank Losses at time t
      atkloss=uaa*(vlatkt(3)-inamb)
      atk55=uaa*(55-inamb)
C Tank Capacitance a) by Bulk Volume,
C b) by nodes (MCp dT/dt, integration)
      CALL RhocpW(vlatkt(2),cp,den)
      blkcapa=volumea*cp*den*(vlatkt(3)-vlatkt(1))/(2.0*dt)
      ndlcapa=cp*den*((tbot(3)-tbot(1))+(t2(3)-t2(1))+
+ (t3(3)-t3(1))+(t4(3)-t4(1))+(t5(3)-t5(1))+(t6(3)-t6(1))+
+ (t7(3)-t7(1))*volabot+(ttop(3)-ttop(1))*volatop)/(2.0*dt)
C
C Calculations to be performed during draws only
      IF(dt.eq.15.0) THEN
C Draw energy across the collector
          t=(bkcol(3)+incol(3))/2.0
          CALL RhocpW(t,cp,den)
          drwcol=drwflw*den*cp*(bkcol(3)-incol(3))
C Draw energy out of auxiliary tank
          t=(drwt+inaux)/2.0
          CALL RhocpW(t,cp,den)
          drwaux=drwflw*den*cp*(drwt-inaux)
C Total draw energy
          t=(drwt+supt)/2.0
          CALL RhocpW(t,cp,den)
          soldrw=drwflw*den*cp*(drwt-supt)
C Additional Draw energy if tank were kept at 55C
          t=(55+drwt)/2.0
          CALL RhocpW(t,cp,den)
          drw55=drwflw*den*cp*(55-drwt)
C Used for SPF-TRNSYS, Energy across tank
          t=(intnk+bktnk)/2.0
          CALL RhocpW(t,cp,den)
          drwtank=drwflw*den*cp*(intnk-bktnk)
C Solar Savings Fraction - Thermal
          ssfthr=soldrw/(soldrw+drw55+atk55-atkloss)
C Solar Savings Fraction - TRNSYS
          ssftrn=drwtank/(drwtank+drw55+atk55-atkloss)

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```

ENDIF
C Write the output to the appropriate files
IF(dt.eq.15.0) THEN
  IF(iflwNEG.eq.1) THEN
    WRITE(iunit2,*)''
  ENDIF
  WRITE(iunit2,44) soltnk, qu(3), pmpow, vlatkt(3), atkloss,
+   atk55, blkcapa, ndlcapa, drwcol, drwaux, soldrw, drw55,
+   drwtmk, ssfthr, ssftrn, coldt(3), coleff(3), frul, frta, rltim
ELSE
  WRITE(iunit1,43) soltnk, qu(3), pmpow, vlatkt(3), atkloss,
+   atk55, blkcapa, ndlcapa, coldt(3), coleff(3), frul, frta, rltim
ENDIF
43 FORMAT(12(E13.6,1X,', '),F7.3)
44 FORMAT(19(E13.6,1X,', '),F7.3)
C
RETURN
END

C
C
C Subroutine to do energy calculations for Thermo system
SUBROUTINE ENERGYT(var3,tcnt,dt,iunit1,iunit2,rltim,iflwTH)
C
  REAL var3(13,8,3),dt,rltim
  REAL ttop(2,3),t7(2,3),t6(2,3),t5(2,3),outamb(3)
  REAL t4(2,3),t3(2,3),t2(2,3),tbot(2,3),pytilt(3)
  REAL vlmtkt(3),vlatkt(3),coleff(3),qu(3),coldt(3)
  REAL pmpflw(3),intnk,bktnk,incol(3),bkcol(3),delp,inaux
  REAL wintnk,wbktnk,convflw,frul,frta,acnet
  REAL volmbot,volmtop,uam,uaa,volabot,volatop
  REAL volumem,volumea,inamb,drwflw,drwt,supt,t
  REAL soltnk,pmpow,solgly,mtkloss,atkloss,mtk55
  REAL atk55,blkcapm,ndlcapm,blkcapa,ndlcapa,solaux
  REAL drwaux,soldrw,drw55,ssfthr,ssftrn
  REAL effgly,effwat
  INTEGER it,itpl,itml,tcnt,i,j,k,k1,k2,k3,iunit1,iunit2,iflwTH
C Sets t-1, t, and t+1 from main program
  it=tcnt
  itpl=tcnt+1
  itml=tcnt-1
  IF(itpl.eq.4) THEN
    itpl=1
  ENDIF
  IF(itml.eq.0) THEN
    itml=3
  ENDIF
C Sets the system variables
C Thermo System
  intnk=var3(10,2,it)
  bktnk=var3(10,1,it)
  delp=var3(13,3,it)
  inaux=var3(10,5,it)
  wintnk=var3(10,3,it)
  wbktnk=var3(10,4,it)
  convflw=var3(13,2,it)
  volmbot=0.0265
  volmtop=0.0281
  i=11
  j=12

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```

    uam=5.36
    uaa=2.06
    acnet=5.67
C The next two do loops initialize the tank
C temperatures & the collector efficiency
C variables at (k1=1)t-1, (k1=2)t, (k1=3)t+1.
C The outside loop determines (k3=i) Main tank
C and (k3=j) Auxiliary tank.
    DO 10 k1=1,3
        IF(k1.eq.1)THEN
            k2=itp1
        ELSEIF(k1.eq.2)THEN
            k2=itm1
        ELSE
            k2=it
        ENDIF
        pmpflw(k1)=var3(13,1,k2)
        pytilt(k1)=var3(2,4,k2)
        outamb(k1)=var3(1,4,k2)
        incol(k1)=var3(10,6,k2)
        bkcol(k1)=var3(10,7,k2)
        DO 20 k=1,2
            IF(k.eq.1)THEN
                k3=i
            ELSE
                k3=j
            ENDIF
            ttop(k,k1)=var3(k3,8,k2)
            t7(k,k1)=var3(k3,7,k2)
            t6(k,k1)=var3(k3,6,k2)
            t5(k,k1)=var3(k3,5,k2)
            t4(k,k1)=var3(k3,4,k2)
            t3(k,k1)=var3(k3,3,k2)
            t2(k,k1)=var3(k3,2,k2)
            tbot(k,k1)=var3(k3,1,k2)
        20    CONTINUE
    10    CONTINUE
C Non-system specific variables
    volabot=0.01858
    volatop=0.009292
    volumem=volmbot*7+volmtop
    volumea=volabot*7+volatop
    inamb=var3(1,1,it)
    drwflw=var3(2,1,it)
    drwt=var3(1,3,it)
    supt=var3(1,2,it)
C Starting Calculations
C Solar into Storage Tank (Thermo: Water across HX)
    t=(wintnk+wbktnk)/2.0
    CALL RhocpW(t,cp,den)
    soltnk=convflw*cp*(wintnk-wbktnk)
C Pumping Power
    pmpow=pmpflw(3)*delp
C Thermo Only: Glycol across Heat Exchanger
    t=(intnk+bktnk)/2.0
    CALL RhocpG(t,cp,den)
    solgly=pmpflw(3)*cp*den*(intnk-bktnk)
C Volume Average Tank Temperatures at t-1, t, t+1
    DO 30 k=1,3

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      vlmkt(k) = ((tbot(1,k)+t2(1,k)+t3(1,k)+t4(1,k)+
+      t5(1,k)+t6(1,k)+t7(1,k))*volmbot+
+      ttop(1,k)*volmtop)/volumem
      vlatkt(k) = ((tbot(2,k)+t2(2,k)+t3(2,k)+t4(2,k)+
+      t5(2,k)+t6(2,k)+t7(2,k))*volabot+
+      ttop(2,k)*volatop)/volumea
C Solar across Collectors
      t=(incol(k)+bkcol(k))/2.0
      CALL RhocpG(t,cp,den)
      qu(k)=pmpflw(k)*cp*den*(bkcol(k)-incol(k))
      coleff(k)=qu(k)/(acnet*pytilt(k))
      coldt(k)=(incol(k)-outamb(k))/pytilt(k)
30 CONTINUE
C Collector Efficiency slope (FrUL) and intercept (FrTa)
      frul=-1*((coleff(3)-coleff(2))/(coldt(3)-coldt(2)))
      frta=coleff(3)+frul*coldt(3)
C Tank Losses at time t
      mtkloss=uam*(vlmkt(3)-inamb)
      atkloss=uaa*(vlatkt(3)-inamb)
      mtk55=uam*(55-inamb)
      atk55=uaa*(55-inamb)
C Tank Capacitance a) by Bulk Volume,
C b) by nodes (MCp dT/dt, integration)
      CALL RhocpW(vlmkt(2),cp,den)
      blkcapm=volumem*cp*den*(vlmkt(3)-vlmkt(1))/(2.0*dt)
      ndlcapm=cp*den*((tbot(1,3)-tbot(1,1))+t2(1,3)-t2(1,1))+
+      t3(1,3)-t3(1,1))+t4(1,3)-t4(1,1))+t5(1,3)-t5(1,1))+
+      t6(1,3)-t6(1,1))+t7(1,3)-t7(1,1))*volmbot+ttop(1,3)-
+      ttop(1,1))*volmtop)/(2.0*dt)
      CALL RhocpW(vlatkt(2),cp,den)
      blkcapa=volumea*cp*den*(vlatkt(3)-vlatkt(1))/(2.0*dt)
      ndlcapa=cp*den*((tbot(2,3)-tbot(2,1))+t2(2,3)-t2(2,1))+
+      t3(2,3)-t3(2,1))+t4(2,3)-t4(2,1))+t5(2,3)-t5(2,1))+
+      t6(2,3)-t6(2,1))+t7(2,3)-t7(2,1))*volabot+ttop(2,3)-
+      ttop(2,1))*volatop)/(2.0*dt)
C Calculations for Heat Exchanger performance
C Effectiveness using glycol flow for Q
      t=(wintnk+wbktnk)/2.0
      CALL RhocpW(t,cp,den)
      effgly=solgly/(convflw*cp*(intnk-wbktnk))
C Effectiveness using water flow for Q
      effwat=(wintnk-wbktnk)/(intnk-wbktnk)
C
C Calculations to be performed during draws only
      IF(dt.eq.15.0) THEN
C Draw energy into auxiliary tank
      t=(inaux+supt)/2.0
      CALL RhocpW(t,cp,den)
      solaux=drwflw*den*cp*(inaux-supt)
C Draw energy out of auxiliary tank
      t=(drwt+inaux)/2.0
      CALL RhocpW(t,cp,den)
      drwau=drwflw*den*cp*(drwt-inaux)
C Total draw energy
      t=(drwt+supt)/2.0
      CALL RhocpW(t,cp,den)
      soldrw=drwflw*den*cp*(drwt-supt)
C Additional Draw energy if tank were kept at 55C
      t=(55+drwt)/2.0

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      CALL RhocpW(t, cp, den)
      drw55=drwflw*den*cp*(55-drwt)
C Solar Savings Fraction - Thermal
      ssfthr=soldrw/(soldrw+drw55+atk55-atkloss)
C Solar Savings Fraction - TRNSYS
      ssftrn=solaux/(solaux+drw55+atk55-atkloss)
      ENDIF
C Write the output to the appropriate files
      IF (dt.eq.15.0) THEN
        IF (iflwTH.eq.1) THEN
          WRITE(iunit2,*)''
        ENDIF
        WRITE(iunit2,47) soltnk, qu(3), pmppow, solgly, vlmtkt(3), vlatkt(3),
+       mtkloss, atkloss, mtk55, atk55, blkcapm, ndlcapm, blkcapa, ndlcapa,
+       effgly, effwat, solaux, drwaux, soldrw, drw55, ssfthr, ssftrn,
+       coldt(3), coleff(3), frul, frta, rltim
      ELSE
        WRITE(iunit1,46) soltnk, qu(3), pmppow, solgly, vlmtkt(3), vlatkt(3),
+       mtkloss, atkloss, mtk55, atk55, blkcapm, ndlcapm, blkcapa, ndlcapa,
+       effgly, effwat, coldt(3), coleff(3), frul, frta, rltim
      ENDIF
46 FORMAT(20(E13.6,1X,', '),F7.3)
47 FORMAT(26(E13.6,1X,', '),F7.3)
      RETURN
      END

```

C
C

```

      SUBROUTINE RhocpW(t, cp, den)
C Used to determine water properties
C Cp=J/kg-K, Den=Kg/m^3
      REAL t, cp, den
      cp=-5.9028e-5*t**3+0.025149*t**2-1.8865*t+4215.7
      den=-0.0030357*t**2-0.11905*t+1003.3
      RETURN
      END

```

C
C

```

      SUBROUTINE RhocpG(t, cp, den)
C Used to determine glycol properties
C Cp=J/kg-K, Den=Kg/m^3
      REAL t, cp, den
      cp=-8.9729e-5*t**3+0.027076*t**2-0.99634*t+3681.6
      den=-0.0015693*t**2-0.52002*t+1044.6
      RETURN
      END

```

PROGRAM LEASTSQ

```

C      To solve a set of a fit to data using linear regression.
      PARAMETER (NDATA1=1350, NPAR1=10, SIG=1.0, TOL=1.E-5)
      REAL EFF, TI, TA, EXTRA, ZEN, HORZ, BEAMH, DIFTH, THET, CTHET, RHOG, PI
      REAL CHIED, CHIEG, CHI, TAU, DTR, SIG, CHISQ, TOL, TIME, SE
      REAL AFUNC(NDATA1, NPAR1), Y(NDATA1), A(NPAR1), V(NPAR1, NPAR1)
      REAL U(NDATA1, NPAR1), W(NPAR1), CVM(NPAR1, NPAR1), B(NDATA1)
      REAL PRED(NDATA1), RES(NDATA1), TILT, FRUL, AVEPRE, AVEY, SUMP, SUMY
      REAL AZI, BMTILT, DIFTILT, TOUT, VDOT, WNDSPD, CTHET1, RAB(NPAR1, NPAR1)
      REAL SUMP, SUMRA, SUMR2, TLTM, TLTM1(NDATA1), OPRE(NDATA1)

```

```

REAL TI1 (NDATA1),TA1 (NDATA1),SUMOPRE,QSRCC (NDATA1),QSRCC1,C
INTEGER NDATA,NPAR,MP,NP,I,J,K,ICNT,IT,NDATA1,INUM,IBETA,NPAR1
INTEGER NUMPAR
CHARACTER JUNK*80,INAME*14,ITHET*5,IRAD*5

C
C Opening the input and output data files
OPEN(UNIT=1,FILE='RAD.OUT',ERR=700,STATUS='OLD')
OPEN(UNIT=3,FILE='RAD2.OUT',ERR=700,STATUS='OLD')
OPEN(UNIT=2,FILE='PAR.DAT',ERR=700,STATUS='NEW')
*OPEN(UNIT=4,FILE='OUT.DAT',ERR=700,STATUS='NEW')
C
PI=3.1415926536
DTR=PI/180.
*TAUA=1.01*0.9 (Tau) *0.95 (alpha)
TAUA=0.864
RHOG=0.5

C
C This portion of the program reads in the data from TRNSY & EXP.
C data and calculates the 'X' portions of the parameter equation
PRINT*,'Input the number of the correlation you would like.'
PRINT*,'HWB=1, GT=2, IbId(tlt)=3, Reindl=4, '
PRINT*,'FrUl (aV+b)=5, Fr&FrUl (av+b)=6'
READ*,INUM
IF (INUM.EQ.1) INAME='HWB'
IF (INUM.EQ.2) INAME='MEAS.'
IF (INUM.EQ.3) INAME='IbId(tlt)'
IF (INUM.EQ.4) INAME='Reindl'
IF (INUM.EQ.5) INAME='FrUl (av+b)'
IF (INUM.EQ.6) INAME='Fr&FrUl (av+b)'
PRINT*,'Input whether to use theta or beta in (taualpha)ave.'
PRINT*,'Theta=1, Beta=2 cos4=3'
READ*,IBETA
IF (IBETA.EQ.1) ITHET='Theta'
IF (IBETA.EQ.2) ITHET='Beta'
IF (IBETA.EQ.3) ITHET='Cos4'
PRINT*,'Input the number of parameters, HWB=2, 3, 4'
PRINT*,'cos4=6, FrUl=3, Fr&FrUl=4'
READ*,NUMPAR
PRINT*,'Input the percentage of radiation data to use,'
PRINT*,'80%=790.0, 90%=395.0, 100%=0.0'
READ*,RADLIM
IF (RADLIM.EQ.790.0) IRAD='80%'
IF (RADLIM.EQ.395.0) IRAD='90%'
IF (RADLIM.EQ.0.0) IRAD='100%'
PRINT*,'Input FrUl from data if using, enter 1.0 if not'
READ*,FRUL

C
IT=0
NPAR=0
READ(1,105) JUNK
READ(3,105) JUNK
DO 10 ICNT=1,NDATA1
READ(1,100,END=1,ERR=700) TIME,EXTRA,ZEN,TILT,HORZ,DIFFH,
+ THET,TI,TA,EFF,TLTM
1 READ(3,101,END=15,ERR=700) TIME,AZI,BMTLT,DIFTLT,TOUT,VOUT,
+ WNDSPD

C
C Only including the top 80% of Radiation data: >= 790 kJ/hr m^2
C Top 90%: Radiation data >= 790/2 kJ/hr m^2, All >= 0 kJ/hr m^2

```

```

      IF ( (TILT.GE.RADLIM) .AND. (EFF.GT.0.0) ) THEN
        IT=IT+1
        TLTM1 (IT)=TLTM
        TI1 (IT)=TI
        TA1 (IT)=TA
        C=COS (THET*DTR)
        QSRCC1=5.9* (TILT/3.6) *0.6367* (1.-0.1559471* ((1./C)-1.))
        QSRCC (IT)=QSRCC1-5.9*4.536* (TI-TA)
      *
      WRITE (4,*) ICNT, IT, TLTM, TLTM1 (IT)
    C
    C HWB Correlation, Linear, 2 parameters, FrTa & FrU1 (W/m^2 C)
      IF (INUM.EQ.1) THEN
        IF (FRUL.GT.1.0) THEN
          NPAR=1
          AFUNC (IT,1)=5.9* (TILT/3.6)
        ELSE
          NPAR=2
          AFUNC (IT,1)=5.9* (TILT/3.6)
          AFUNC (IT,2)=-5.9*FRUL* (TI-TA)
        ENDIF
      C
      C Just separating (TAUALP)ave into 3 Components
      C Using Tilt data only, 3 & 4 parameters: linear theta, Fr, Frb0,
      C FrU1 (W/m^2 C), quadratic theta, add Frbl
      ELSEIF ( (INUM.GE.2) .AND. (INUM.LT.5) ) THEN
      C
        IF ( (IBETA.EQ.1) .OR. (IBETA.EQ.3) ) THEN
          CTHET=COS (THET*DTR)
        ELSEIF (IBETA.EQ.2) THEN
          CTHET=COS (ASIN ( (1./1.526) *SIN (THET*DTR) ))
        ENDIF
        IF (IBETA.EQ.3) THEN
          CHI=CTHET
          CHIED=COS (69.4*DTR)
          CHIEG=COS (56.5*DTR)
        ELSE
          CHI= (1./CTHET) -1.
          CHIED= (1./COS (69.4*DTR) ) -1.
          CHIEG= (1./COS (56.5*DTR) ) -1.
        ENDIF
      C
      IF ( (INUM.EQ.2) .AND. (NUPAR.EQ.3) ) THEN
        IF (FRUL.GT.1.0) THEN
          NPAR=2
          AFUNC (IT,1)=5.9* (TILT/3.6) *TAUA
          AFUNC (IT,2)=-5.9* (TILT/3.6) *TAUA*CHI
        ELSE
          NPAR=3
          AFUNC (IT,1)=5.9* (TILT/3.6) *TAUA
          AFUNC (IT,2)=-5.9* (TILT/3.6) *TAUA*CHI
          AFUNC (IT,3)=-5.9*FRUL* (TI-TA)
        ENDIF
      ELSEIF ( (INUM.EQ.2) .AND. (NUPAR.EQ.4) ) THEN
        IF (FRUL.GT.1.0) THEN
          NPAR=3
          AFUNC (IT,1)=5.9* (TILT/3.6) *TAUA
          AFUNC (IT,2)=-5.9* (TILT/3.6) *TAUA*CHI
          AFUNC (IT,3)=-5.9* (TILT/3.6) *TAUA*CHI**2
        ELSE

```

```

      NPAR=4
      AFUNC(IT,1)=5.9*(TILT/3.6)*TAUA
      AFUNC(IT,2)=-5.9*(TILT/3.6)*TAUA*CHI
      AFUNC(IT,3)=-5.9*(TILT/3.6)*TAUA*CHI**2
      AFUNC(IT,4)=-5.9*FRUL*(TI-TA)
    ENDIF
  ELSEIF ((INUM.EQ.2).AND.(NUPAR.EQ.6)) THEN
    IF (FRUL.GT.1.0) THEN
      NPAR=5
      AFUNC(IT,1)=5.9*(TILT/3.6)*TAUA
      AFUNC(IT,2)=5.9*(TILT/3.6)*TAUA*CHI
      AFUNC(IT,3)=5.9*(TILT/3.6)*TAUA*CHI**2
      AFUNC(IT,4)=5.9*(TILT/3.6)*TAUA*CHI**3
      AFUNC(IT,5)=5.9*(TILT/3.6)*TAUA*CHI**4
    ELSE
      NPAR=6
      AFUNC(IT,1)=5.9*(TILT/3.6)*TAUA
      AFUNC(IT,2)=5.9*(TILT/3.6)*TAUA*CHI
      AFUNC(IT,3)=5.9*(TILT/3.6)*TAUA*CHI**2
      AFUNC(IT,4)=5.9*(TILT/3.6)*TAUA*CHI**3
      AFUNC(IT,5)=5.9*(TILT/3.6)*TAUA*CHI**4
      AFUNC(IT,6)=-5.9*FRUL*(TI-TA)
    ENDIF
  C
  C Using TRNSYS outputs for Ibeam(tilt) and Idiff(tilt)
  C 3 parameters: Fr, Frb0, Frul(W/m^2 C), 4: add Frb1
  ELSEIF ((INUM.EQ.3).AND.(NUPAR.EQ.3)) THEN
    IF (FRUL.GT.1.0) THEN
      NPAR=2
      AFUNC(IT,1)=5.9*((BMTLT+DIFTLT)/3.6)*TAUA
      AFUNC(IT,2)=-5.9*TAUA*(BMTLT*CHI+DIFTLT*CHIED)/3.6
    ELSE
      NPAR=3
      AFUNC(IT,1)=5.9*((BMTLT+DIFTLT)/3.6)*TAUA
      AFUNC(IT,2)=-5.9*TAUA*(BMTLT*CHI+DIFTLT*CHIED)/3.6
      AFUNC(IT,3)=-5.9*FRUL*(TI-TA)
    ENDIF
  ELSEIF ((INUM.EQ.3).AND.(NUPAR.EQ.4)) THEN
    IF (FRUL.GT.1.0) THEN
      NPAR=3
      AFUNC(IT,1)=5.9*((BMTLT+DIFTLT)/3.6)*TAUA
      AFUNC(IT,2)=-1*TAUA*(BMTLT*CHI+DIFTLT*CHIED)/3.6
      AFUNC(IT,3)=-1*TAUA*(BMTLT*CHI**2+DIFTLT*CHIED**2)/3.6
    ELSE
      NPAR=4
      AFUNC(IT,1)=5.9*((BMTLT+DIFTLT)/3.6)*TAUA
      AFUNC(IT,2)=-5.9*TAUA*(BMTLT*CHI+DIFTLT*CHIED)/3.6
      AFUNC(IT,3)=-5.9*TAUA*(BMTLT*CHI**2+DIFTLT*CHIED**2)/3.6
      AFUNC(IT,4)=-5.9*FRUL*(TI-TA)
    ENDIF
  ELSEIF ((INUM.EQ.3).AND.(NUPAR.EQ.6)) THEN
    IF (FRUL.GT.1.0) THEN
      NPAR=5
      AFUNC(IT,1)=5.9*((BMTLT+DIFTLT)/3.6)*TAUA
      AFUNC(IT,2)=5.9*TAUA*(BMTLT*CHI+DIFTLT*CHIED)/3.6
      AFUNC(IT,3)=5.9*TAUA*(BMTLT*CHI**2+DIFTLT*CHIED**2)/3.6
      AFUNC(IT,4)=5.9*TAUA*(BMTLT*CHI**3+DIFTLT*CHIED**3)/3.6
      AFUNC(IT,5)=5.9*TAUA*(BMTLT*CHI**4+DIFTLT*CHIED**4)/3.6
    ELSE

```

```

      NPAR=6
      AFUNC(IT,1)=5.9*( (BMTLT+DIFTLT)/3.6)*TAUA
      AFUNC(IT,2)=5.9*TAUA*(BMTLT*CHI+DIFTLT*CHIED)/3.6
      AFUNC(IT,3)=5.9*TAUA*(BMTLT*CHI**2+DIFTLT*CHIED**2)/3.6
      AFUNC(IT,4)=5.9*TAUA*(BMTLT*CHI**3+DIFTLT*CHIED**3)/3.6
      AFUNC(IT,5)=5.9*TAUA*(BMTLT*CHI**4+DIFTLT*CHIED**4)/3.6
      AFUNC(IT,6)=-5.9*FRUL*(TI-TA)

```

```

    ENDIF

```

C

C Reindl Correlation, linear theta, 3 parameters, Fr, Frb0,

C FrU1 (W/m^2 C), quadratic theta, 4 parameters, add Frb1

```

      ELSEIF (INUM.GT.3) THEN

```

```

        CTHET1=COS(THET*DTR)
        BETA=45.*DTR
        BEAMH=HORZ-DIFFH
        AI=BEAMH/EXTRA
        F=(BEAMH/HORZ)**0.5
        F1=(BEAMH+DIFFH*AI)*CTHET1/(COS(ZEN*DTR))
        F2=DIFFH*(1.-AI)*((1.+COS(BETA))/2.)*(1.+F*(SIN(BETA/2.))**3)
        F3=HORZ*RHO*(1.-COS(BETA))/2.
        IF ((INUM.EQ.4).AND.(NUPAR.EQ.3)) THEN

```

```

            IF (FRUL.GT.1.0) THEN

```

```

                NPAR=2

```

```

                AFUNC(IT,1)=5.9*TAUA*(F1+F2+F3)/3.6

```

```

                AFUNC(IT,2)=-5.9*(TAUA*(F1*CHI+F2*CHIED+F3*CHIEG)/3.6)

```

```

            ELSE

```

```

                NPAR=3

```

```

                AFUNC(IT,1)=5.9*TAUA*(F1+F2+F3)/3.6

```

```

                AFUNC(IT,2)=-5.9*(TAUA*(F1*CHI+F2*CHIED+F3*CHIEG)/3.6)

```

```

                AFUNC(IT,3)=-5.9*FRUL*(TI-TA)

```

```

            ENDIF

```

```

      ELSEIF ((INUM.EQ.4).AND.(NUPAR.EQ.4)) THEN

```

```

        IF (FRUL.GT.1.0) THEN

```

```

            NPAR=3

```

```

            AFUNC(IT,1)=5.9*TAUA*(F1+F2+F3)/3.6

```

```

            AFUNC(IT,2)=-5.9*(TAUA*(F1*CHI+F2*CHIED+F3*CHIEG)/3.6)

```

```

            AFUNC(IT,3)=-5.9*(TAUA*(F1*CHI**2+F2*CHIED**2+F3*CHIEG**2)/3.6)

```

```

        ELSE

```

```

            NPAR=4

```

```

            AFUNC(IT,1)=5.9*TAUA*(F1+F2+F3)/3.6

```

```

            AFUNC(IT,2)=-5.9*(TAUA*(F1*CHI+F2*CHIED+F3*CHIEG)/3.6)

```

```

            AFUNC(IT,3)=-5.9*(TAUA*(F1*CHI**2+F2*CHIED**2+F3*CHIEG**2)/3.6)

```

```

            AFUNC(IT,4)=-5.9*FRUL*(TI-TA)

```

```

        ENDIF

```

```

      ELSEIF ((INUM.EQ.4).AND.(NUPAR.EQ.6)) THEN

```

```

        IF (FRUL.GT.1.0) THEN

```

```

            NPAR=5

```

```

            AFUNC(IT,1)=5.9*TAUA*(F1+F2+F3)/3.6

```

```

            AFUNC(IT,2)=5.9*(TAUA*(F1*CHI+F2*CHIED+F3*CHIEG)/3.6)

```

```

            AFUNC(IT,3)=5.9*(TAUA*(F1*CHI**2+F2*CHIED**2+F3*CHIEG**2)/3.6)

```

```

            AFUNC(IT,4)=5.9*(TAUA*(F1*CHI**3+F2*CHIED**3+F3*CHIEG**3)/3.6)

```

```

            AFUNC(IT,5)=5.9*(TAUA*(F1*CHI**4+F2*CHIED**4+F3*CHIEG**4)/3.6)

```

```

        ELSE

```

```

            NPAR=6

```

```

            AFUNC(IT,1)=5.9*TAUA*(F1+F2+F3)/3.6

```

```

            AFUNC(IT,2)=5.9*(TAUA*(F1*CHI+F2*CHIED+F3*CHIEG)/3.6)

```

```

            AFUNC(IT,3)=5.9*(TAUA*(F1*CHI**2+F2*CHIED**2+F3*CHIEG**2)/3.6)

```

```

            AFUNC(IT,4)=5.9*(TAUA*(F1*CHI**3+F2*CHIED**3+F3*CHIEG**3)/3.6)

```

```

            AFUNC(IT,5)=5.9*(TAUA*(F1*CHI**4+F2*CHIED**4+F3*CHIEG**4)/3.6)

```

```

                                AFUNC(IT, 6)=-5.9*FRUL*(TI-TA)
                                ENDIF
                                ENDIF
                                ENDIF
C
C Breaking down Frul into a*wndspd+b
                                ELSEIF (INUM.EQ.5) THEN
                                    NPAR=3
                                    AFUNC(IT, 1)=5.9*TILT/3.6
                                    AFUNC(IT, 2)=-5.9*WNDSPD*(TI-TA)
                                    AFUNC(IT, 3)=-5.9*(TI-TA)
                                ELSEIF (INUM.EQ.6) THEN
                                    NPAR=4
                                    AFUNC(IT, 1)=5.9*TILT/3.6
                                    AFUNC(IT, 2)=5.9*(TILT/3.6)*WNDSPD
                                    AFUNC(IT, 3)=-5.9*WNDSPD*(TI-TA)
                                    AFUNC(IT, 4)=-5.9*(TI-TA)
                                ENDIF
C
C Y values equal the efficiency
                                IF (FRUL.GT.1.0) THEN
*                                    Y(IT)=VOUT*1006*3870*(TOUT-TI)+5.9*FRUL*(TI-TA)
*                                    Y(IT)=EFF*5.9*(TILT/3.6)+5.9*FRUL*(TI-TA)
                                ELSE
*                                    Y(IT)=VOUT*1006*3870*(TOUT-TI)
*                                    Y(IT)=EFF*5.9*(TILT/3.6)
                                ENDIF
                                ENDIF
10 CONTINUE
C
C This portion calculates the parameters using Numerical Recipes
C programs
15 CONTINUE
    MP=NDATA1
    NDATA=IT
    NP=NPAR1
    MA=NPAR
    DO 17 I=1,NDATA
        DO 20 J=1,MA
            U(I,J)=AFUNC(I,J)*(1./SIG)
20        CONTINUE
            B(I)=Y(I)*(1./SIG)
17 CONTINUE
    CALL SVDCMP(U,NDATA,MA,MP,NP,W,V)
    WMAX=0.0
    DO 30 J=1,MA
        IF (W(J).GT.WMAX) THEN
            WMAX=W(J)
        ENDIF
30 CONTINUE
    THRESH=TOL*WMAX
    DO 40 J=1,MA
        IF (W(J).LT.THRESH) THEN
            W(J)=0.0
        ENDIF
40 CONTINUE
    CALL SVBKS(B,U,W,V,NDATA,MA,MP,NP,B,A)
    CHISQ=0.0
    SUMPRES=0.0

```

```

SUMY=0.0
SUMRP=0.0
SUMRA=0.0
SUMR2=0.0
SUMQPRE=0.0
DO 50 I=1,NDATA
    SUM=0.0
    DO 60 J=1,MA
        SUM=SUM+A(J)*AFUNC(I,J)
60    CONTINUE
    PRED(I)=SUM
    RES(I)=Y(I)-SUM
    IF (FRUL.GT.1.0) THEN
        QPRE(I)=PRED(I)-5.9*FRUL*(TI1(I)-TA1(I))
        SUMQPRE=SUMQPRE+QPRE(I)
    ENDIF
    CHISQ=CHISQ+((Y(I)-SUM)/SIG)**2
    SUMPRE=SUMPRE+PRED(I)
    SUMY=SUMY+Y(I)
50 CONTINUE
AVEPRE=SUMPRE/NDATA
AVEY=SUMY/NDATA
DO 61 I=1,NDATA
    SUMRP=SUMRP+(PRED(I)-AVEY)**2
    SUMRA=SUMRA+(Y(I)-AVEY)**2
61 CONTINUE
SUMR2=SUMRP/SUMRA
SE=(CHISQ/NDATA)**0.5
CALL SVDVAR(V,NPAR,NP,W,CVM,NP)
C Calculate the correlation matrix:
DO 65 I=1,NPAR
    DO 70 J=1,NPAR
        RAB(I,J)=CVM(I,J)/((CVM(I,I)**0.5)*(CVM(J,J)**0.5))
70    CONTINUE
65 CONTINUE
C Write the output to a file
C Write The Parameters:
WRITE(2,*) 'A least squares fit is performed on the measured effici
*   ency and collector inlet and ambient temperatures. The radiation
*   data is output from TRNSYS using measured horizontal data. No nigh
*   t time data is included in the fit.'
WRITE(2,*) ''
WRITE(2,*) 'Since Sigma is unknown and assumed to be 1, the chi-squ
*   are means nothing. The standard error in estimated y is sqrt(chisq
*   uare/#points). The standard error in each parameter is the sqrt of
*   each parameter number in the diagonal of the covariance matrix'
WRITE(2,*) ''
WRITE(2,*) 'Corr: ',INAME,' Using ',NUNPAR,' Parameters, ',ITHET,
*   ' and ',IRAD,' of the data; FrU1=',FRUL,' (if 1, its a parameter)'
WRITE(2,*) ''
WRITE(2,*) 'Parameters:'
DO 80 I=1,NPAR
    WRITE(2,110) I,A(I),CVM(I,I)**0.5
80 CONTINUE
WRITE(2,*) ''
WRITE(2,112) (A(I),I=1,NPAR)
WRITE(2,*) ''
WRITE(2,*) 'Chi-squared =',CHISQ
WRITE(2,*) 'Standard error =',SE

```

```

WRITE(2,*)'r-squared =',SUMR2
WRITE(2,*)'Average measured Y =',AVEY
WRITE(2,*)'Average predicted Y =',AVEPRE
WRITE(2,*)'Total measured Y: Quse =',SUMY*300.
WRITE(2,*)'Total predicted Y: Quse =',SUMPRE*300.
IF(FRUL.GT.1.0)THEN
    WRITE(2,*)'Total predicted Y2: Quse =',SUMOPRE*300.
ENDIF
C Write the covariance matrix:
WRITE(2,*)''
WRITE(2,*)'The Covariance Matrix:'
WRITE(2,130) (I, I=1,NPAR)
DO 135 I=1,NPAR
    WRITE(2,140) I, (CVM(I,J),J=1,NPAR)
135 CONTINUE
C Write the correlation matrix:
WRITE(2,*)''
WRITE(2,*)'The Correlation Matrix:'
WRITE(2,130) (I, I=1,NPAR)
DO 136 I=1,NPAR
    WRITE(2,140) I, (RAB(I,J),J=1,NPAR)
136 CONTINUE
C Write the predicted y's and residuals:
WRITE(2,*)''
WRITE(2,*)'The Predicted Y Values and Residuals:' WRITE(2,115)
DO 120 i=1,NDATA
    IF (FRUL.GT.1.0) THEN
        WRITE(2,125) TLTM1(I)*5.9/3.6,I,Y(I),PRED(I),RES(I),QPRE(I)
    ELSE
        WRITE(2,125) TLTM1(I)*5.9/3.6,I,Y(I),PRED(I),RES(I),QSRCC(I)
    ENDIF
120 CONTINUE
C Formatting output
100 FORMAT(F10.4,2X,10(E10.3,2X))
101 FORMAT(F10.4,2X,6(E10.3,2X))
105 FORMAT(A80)
110 FORMAT('A(',I1,') = ',F10.5,' +/- ',F10.5)
112 FORMAT('Y = ',F10.5,'+',F10.5,'*F(x1)+',F10.5,'*F(x2)+',
    *F10.5,'*F(x3)+',F10.5,'*F(x4)+',F10.5,'*F(x5)')
115 FORMAT('AcGT',T17,'I',T27,'Y(I)',T40,'PRED',T54,'RES')
125 FORMAT(F12.5,3X,I5,3X,4(F12.5,3X))
130 FORMAT('PAR.',T14,I1,T26,I1,T38,I1,T50,I1,T62,I1,T74,I1)
140 FORMAT(T2,I1,T9,F14.4,T23,F14.4,T35,F14.4,T47,F14.4,T59,F14.4,
    *T71,F14.4)
GOTO 200
700 PRINT*,'Error occurred reading input file' 200 STOP
END

```

SUBROUTINE SVDAMP(A,M,N,MP,NP,W,V)

C Given matrix A, logical dimension MxN and physical dimensions MPxNP. This gives its
C singular value decomposition, A=UWV(transpose)
PARAMETER (NMAX=10)
DIMENSION A(MP,NP),W(NP),V(NP,NP),RV1(NMAX)
IF(M.LT.N)PAUSE 'You must augment A with extra zero rows.'

C Householder reduction to bidiagonal form.

```

G=0.0
SCALE=0.0
ANORM=0.0

```



```

DO 10 I=1,N
  L=I+1
  RV1(I)=SCALE*G
  G=0.0
  S=0.0
  SCALE=0.0
  IF(I.LE.M) THEN
    DO 20 K=I,M
      SCALE=SCALE+ABS(A(K,I))
20    CONTINUE
    IF(SCALE.NE.0.0) THEN
      DO 30 K=I,M
        A(K,I)=A(K,I)/SCALE
        S=S+A(K,I)*A(K,I)
30    CONTINUE
      F=A(I,I)
      G=-SIGN(SQRT(S),F)
      H=F*G-S
      A(I,I)=F-G
      IF(I.NE.N) THEN
        DO 40 J=L,N
          S=0.0
          DO 50 K=I,M
            S=S+A(K,I)*A(K,J)
50        CONTINUE
          F=S/H
          DO 60 K=I,M
            A(K,J)=A(K,J)+F*A(K,I)
60        CONTINUE
40      CONTINUE
      ENDIF
      DO 70 K=I,M
        A(K,I)=SCALE*A(K,I)
70    CONTINUE
      ENDIF
    ENDIF
    W(I)=SCALE*G
    G=0.0
    S=0.0
    SCALE=0.0
    IF((I.LE.M).AND.(I.NE.N)) THEN
      DO 80 K=L,N
        SCALE=SCALE+ABS(A(I,K))
80    CONTINUE
      IF(SCALE.NE.0.0) THEN
        DO 90 K=L,N
          A(I,K)=A(I,K)/SCALE
          S=S+A(I,K)*A(I,K)
90    CONTINUE
          F=A(I,L)
          G=-SIGN(SQRT(S),F)
          H=F*G-S
          A(I,L)=F-G
          DO 100 K=L,N
            RV1(K)=A(I,K)/H
100    CONTINUE
          IF(I.NE.M) THEN
            DO 110 J=L,M
              S=0.0

```

```

DO 120 K=L,N
  S=S+A(J,K)*A(I,K)
120  CONTINUE
DO 130 K=L,N
  A(J,K)=A(J,K)+S*RV1(K)
130  CONTINUE
110  CONTINUE
      ENDIF
      DO 140 K=L,N
        A(I,K)=SCALE*A(I,K)
140  CONTINUE
      ENDIF
    ENDIF
    ANORM=MAX(ANORM, (ABS(W(I))+ABS(RV1(I))))
10  CONTINUE
C Accumulation of right hand transformations.
DO 150 I=N,1,-1
  IF (I.LT.N) THEN
    IF (G.NE.0.0) THEN
      DO 160 J=L,N
        V(J,I)=(A(I,J)/A(I,L))/G
160  CONTINUE
      DO 170 J=L,N
        S=0.0
        DO 180 K=L,N
          S=S+A(I,K)*V(K,J)
180  CONTINUE
        DO 190 K=L,N
          V(K,J)=V(K,J)+S*V(K,I)
190  CONTINUE
170  CONTINUE
      ENDIF
      DO 200 J=L,N
        V(I,J)=0.0
        V(J,I)=0.0
200  CONTINUE
      ENDIF
      V(I,I)=1.0
      G=RV1(I)
      L=I
150  CONTINUE
C Accumulation of left-hand transformations.
DO 210 I=N,1,-1
  L=I+1
  G=W(I)
  IF (I.LT.N) THEN
    DO 220 J=L,N
      A(I,J)=0.0
220  CONTINUE
    ENDIF
    IF (G.NE.0.0) THEN
      G=1.0/G
      IF (I.NE.N) THEN
        DO 230 J=L,N
          S=0.0
          DO 240 K=L,M
            S=S+A(K,I)*A(K,J)
240  CONTINUE
          F=(S/A(I,I))*G

```

```

DO 250 K=I,M
  A(K,J)=A(K,J)+F*A(K,I)
250 CONTINUE
230 CONTINUE
  ENDIF
  DO 260 J=I,M
    A(J,I)=A(J,I)*G
260 CONTINUE
  ELSE
    DO 270 J=I,M
      A(J,I)=0.0
270 CONTINUE
    ENDIF
    A(I,I)=A(I,I)+1.0
210 CONTINUE

```

C Diagonalization of the bidiagonal form. Loop over singular values. Loop over allowed C iterations. Test for splitting: Note that RV1 is always zero.

```

DO 280 K=N,1,-1
  DO 290 ITS=1,1000
    DO 300 L=K,1,-1
      NM=L-1
      IF ( (ABS(RV1(L))+ANORM) .EQ. ANORM) THEN
        GOTO 2
      ENDIF
      IF ( (ABS(W(NM))+ANORM) .EQ. ANORM) THEN
        GOTO 1
      ENDIF
300 CONTINUE
      1 C=0.0
      S=1.0
      DO 310 I=L,K
        F=S*RV1(I)
        RV1(I)=C*RV1(I)
        IF ( (ABS(F)+ANORM) .EQ. ANORM) THEN
          GOTO 2
        ENDIF
        G=W(I)
        H=SQRT(F*F+G*G)
        W(I)=H
        H=1.0/H
        C=(G*H)
        S=-(F*H)
        DO 320 J=1,M
          Y=A(J,NM)
          Z=A(J,I)
          A(J,NM)=(Y*C)+(Z*S)
          A(J,I)=-(Y*S)+(Z*C)
320 CONTINUE
310 CONTINUE
      2 Z=W(K)
      IF (L.EQ.K) THEN
        IF (Z.LT.0.0) THEN
          W(K)=-Z
          DO 330 J=1,N
            V(J,K)=-V(J,K)
330 CONTINUE
          ENDIF
          GOTO 3
        ENDIF

```

```

IF (ITS.EQ.1000)PAUSE 'No convergence in 1000 iterations' X=W(L)
NM=K-1
Y=W(NM)
G=RV1(NM)
H=RV1(K)
F= ((Y-Z)*(Y+Z)+(G-H)*(G+H))/(2.0*H*Y)
G=SQRT(F*F+1.0)
F= ((X-Z)*(X+Z)+H*((Y/(F+SIGN(G,F)))-H))/X C Next QR transformation:
C=1.0
S=1.0
DO 340 J=L,NM
  I=J+1
  G=RV1(I)
  Y=W(I)
  H=S*G
  G=C*G
  Z=SQRT(F*F+H*H)
  RV1(J)=Z
  C=F/Z
  S=H/Z
  F= (X*C)+(G*S)
  G=-(X*S)+(G*C)
  H=Y*S
  Y=Y*C
  DO 350 JJ=1,N
    X=V(JJ,I)
    Z=V(JJ,I)
    V(JJ,J)=(X*C)+(Z*S)
    V(JJ,I)=-(X*S)+(Z*C)
350  CONTINUE
    Z=SQRT(F*F+H*H)
    W(J)=Z
    IF (Z.NE.0.0) THEN
      Z=1.0/Z
      C=F*Z
      S=H*Z
    ENDIF
    F= (C*G)+(S*Y)
    X=-(S*G)+(C*Y)
    DO 360 JJ=1,M
      Y=A(JJ,J)
      Z=A(JJ,I)
      A(JJ,J)=(Y*C)+(Z*S)
      A(JJ,I)=-(Y*S)+(Z*C)
360  CONTINUE
340  CONTINUE
    RV1(L)=0.0
    RV1(K)=F
    W(K)=X
290  CONTINUE
  3  CONTINUE
280 CONTINUE

```

```

RETURN
END

```

SUBROUTINE SVEKSB(U,W,V,M,N,MP,NP,B,X)

C Solves $AX=B$ for a vector X, A is specified by arrays U,W,V as returned by SVDcmp

```

PARAMETER (NMAX=10)
DIMENSION U(MP,NP),W(NP),V(NP,NP),B(MP),X(NP),TMP(NMAX)
DO 10 J=1,N
    S=0.0
    IF (W(J).NE.0.0) THEN
        DO 20 I=1,M
            S=S+U(I,J)*B(I)
20        CONTINUE
        S=S/W(J)
    ENDIF
    TMP(J)=S
10 CONTINUE
DO 30 J=1,N
    S=0.0
    DO 40 JJ=1,N
        S=S+V(J,JJ)*TMP(JJ)
40    CONTINUE
    X(J)=S
30 CONTINUE
RETURN
END

```

SUBROUTINE SVDVAR(V,MA,NP,W,CVM,NCVM)

C To evaluate the covariance matrix for parameters obtained by SVDfit

```

PARAMETER (MMA=10)
DIMENSION V(NP,NP),W(NP),CVM(NCVM,NCVM),WTI(MMA)
DO 10 I=1,MA
    WTI(I)=0.0
    IF (W(I).NE.0.0) THEN
        WTI(I)=1./(W(I)*W(I))
    ENDIF
10 CONTINUE
DO 20 I=1,MA
    DO 30 J=1,I
        SUM=0.0
        DO 40 K=1,MA
            SUM=SUM+V(I,K)*V(J,K)*WTI(K)
40        CONTINUE
        CVM(I,J)=SUM
        CVM(J,I)=SUM
30    CONTINUE
20 CONTINUE
RETURN
END

```

APPENDIX D

$F_R U_L$ Calculation

The collector heat removal factor, F_R , for a flat-plate collector is given as eqn (D.1). The derivation of F_R is presented in Duffie and Beckman [1991]. F_R is a function of the collector fluid flow rate, $\dot{m}$, the collector specific heat, C_p , the collector area, A_c , the collector overall loss coefficient, U_L , and the collector efficiency factor, F' .

$$F_R = \frac{\dot{m} C_p}{A_c U_L} \left[1 - e^{\frac{-A_c U_L F'}{\dot{m} C_p}} \right] \quad (D.1)$$

The collector efficiency factor is given as eqn (D.2) and is a function of the collector overall loss coefficient, U_L , the distance between the collector tubes, W , the collector tube diameter, D , the inside convective coefficient of the fluid in the collector tubes, h_f , and the fin efficiency factor, F .

$$F = \frac{1/U_L}{W \left[\frac{1}{U_L (D + [W - D] F)} + \frac{1}{h_f \pi D} \right]} \quad (D.2)$$

The fin efficiency factor, F , is given as eqn (D.3). Where k_p is the plate conductivity and δ_p is the plate thickness.

$$F = \frac{\tanh[m(W-D)/2]}{m(W-D)/2}, \quad (D.3)$$

$$m = \left(\frac{U_L}{k_p \delta_p} \right)^{0.5}$$

The overall loss coefficient, U_L , is shown in eqn (D.4) and is a function of the top, back and edge losses. The edge losses are negligible and the back loss coefficient is the conductivity of the insulation, k_{ins} , divided by the insulation thickness, L_{ins} .

$$U_L = U_{Top} + U_{Back} + U_{Edge} \approx U_{Top} + U_{Back} = U_{Top} + \frac{k_{ins}}{L_{ins}} \quad (D.4)$$

The top loss coefficient is the most complicated term and includes all of the convective and radiative losses from the plate to the cover and the cover to the ambient air, as given in eqn (D.5)

$$U_{Top} = \left[\frac{1}{h_{wind} + h_{rad_{cov-amb}}} + \frac{1}{h_{conv_{plate-cov}} + h_{rad_{plate-cov}}} \right]^{-1} \quad (D.5)$$

First, the inside convective coefficient for the glycol flowing in the collector tubes is calculated. The following values were used:

$$\begin{aligned} v_{gly} &= 0.18 \text{ m/s} \\ D &= 0.0095 \text{ m} \\ A &= 7.1 \times 10^{-5} \text{ m}^2 \\ k_{gly} &= 0.42 \text{ W/m}^\circ\text{C} \\ C_{p,gly} &= 3819 \text{ J/kg}^\circ\text{C} \\ \rho_{gly} &= 1007 \text{ kg/m}^3 \\ Pr_{gly} &= 14.6 \\ \mu_{gly} &= 1.6 \times 10^{-3} \text{ Ns/m}^2 \\ L_{gly} &= 30.5 \text{ m} \end{aligned}$$

The resulting Reynolds number of 1077 indicates laminar flow. Therefore, the Seider-Tate relation may be used to determine h_f .

$$h_f = \frac{k}{D}(1.86)\left(RePr\frac{D}{L}\right)^{1/3} \approx 140 \text{ W/m}^2\text{°C} \quad (\text{D.6})$$

Next, the convective loss coefficient due to the wind blowing across the collector face is determined. The wind speed varies between 1 and 5 m/s, and the representative L is 5 m, resulting in h_{wind} between 5 and 18 $\text{W/m}^2\text{°C}$.

$$h_{\text{wind}} = \max\left[5, \frac{8.6V^{0.6}}{L^{0.4}}\right], \text{ W/m}^2\text{°C} \quad (\text{D.7})$$

$V = \text{speed, m/s}$
 $L = (\text{structure volume})^{1/3}, \text{ m}$

The remaining loss coefficients from the collector cover to the ambient, and from the collector plate to the cover, are strong functions of the plate, cover, and ambient temperatures. The maximum and minimum plate temperatures, cover temperatures and ambient temperatures used in the following equations are: $T_{\text{plate,max}} = 100^\circ\text{C}$, $T_{\text{plate,min}} = 10^\circ\text{C}$; $T_{\text{cov,max}} = 50^\circ\text{C}$, $T_{\text{cov,min}} = 0^\circ\text{C}$; and $T_{\text{amb,max}} = 10^\circ\text{C}$, $T_{\text{amb,min}} = -10^\circ\text{C}$. The cover emissivity is 0.88 and the plate emissivity is 0.95.

The radiative loss coefficient for the cover to ambient is given in eqn (D.8) and varies between 3 to 6 $\text{W/m}^2\text{°C}$.

$$h_{\text{rad}_{\text{cov-amb}}} = \epsilon_{\text{cov}}\sigma(T_{\text{cov}}^2 + T_{\text{sky}}^2)(T_{\text{cov}} + T_{\text{sky}}) \quad (\text{D.8})$$

The convective loss coefficient for the plate to cover is given in eqn (D.9) and varies between 4 to 3 $\text{W/m}^2\text{°C}$ (the collector slope, β , is 45°).

$$h_{\text{conv}_{\text{plate-cov}}} = \left(\frac{k_{\text{air}}}{L}\right)\left\{1 + 1.44\left[1 - \frac{1708(\sin 1.8\beta)^{1.6}}{(Ra)\cos\beta}\right]\left[1 - \frac{1708}{(Ra)\cos\beta}\right]^{(+)} + \left[\left(\frac{(Ra)\cos\beta}{5830}\right)^{1/3} - 1\right]^{(+)}\right\} \quad (\text{D.9})$$

The radiative loss coefficient for the plate to cover is given in eqn (D.10) and varies between 3.5 to 8 W/m²°C.

$$h_{rad_{plate-cov}} = \sigma \frac{(T_{plate}^2 + T_{cov}^2)(T_{plate} + T_{cov})}{\frac{1}{\epsilon_{plate}} + \frac{1}{\epsilon_{cov}} - 1} \quad (D.10)$$

Using the above determined values for the different variables results in U_L varying from 4 to 8 W/m²°C and F_R varying between 0.66 and 0.49. Therefore, $F_R U_L$ varies between 2.6 and 3.9 W/m²°C. This range encompasses the experimentally determined value of 3 W/m²°C (night time temperatures). This range also encompasses the values for $F_R U_L$ determined in section 6.4.8 (Duffie and Beckman [1991], page 261) using the average day time and night time plate and ambient temperatures.

Reindl Sky Radiation Model

The Reindl model for sky radiation is used to predict the overall incident radiation on the horizontal given that the incident radiation is comprised of three components: beam, diffuse, and sky diffuse. The collector optics, $(\tau\alpha)$, may then also be separated into the same three components, as shown in eqn (D.11). This model is used in the parameter estimation program when determining $F_R(\tau\alpha)_{beam}$, $F_R(\tau\alpha)_{diff}$, and $F_R(\tau\alpha)_{sky}$. The remaining variables in eqn's (D.11) and (D.12) are output from the TRNSYS radiation processor.

$$\begin{aligned}
S = & (I_{beamH} + I_{diffH}A_i)R_{beam}(\tau\alpha)_{beam} + \\
& I_{diffH}(1 - A_i)(\tau\alpha)_{diff}\left[1 + f\left(\sin\left(\frac{\beta}{2}\right)\right)^3\right]\left(\frac{1 + \cos\beta}{2}\right) + \\
& \rho_{gnd}(I_{beamH} + I_{diffH})(\tau\alpha)_{sky}\left(\frac{1 - \cos\beta}{2}\right)
\end{aligned} \tag{D.11}$$

$$R_{beam} = \frac{\cos\theta}{\cos\theta_z}, \quad A_i = \frac{I_{beamH}}{I_0}, \quad f = \sqrt{\frac{I_{beamH}}{(I_{beamH} + I_{diffH})}} \tag{D.12}$$