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MeteORoLOGICAL EFFECTS ON THERMAL PERFORMANCE

TESTS OF FLAT-PLATE SOLAR COLLECTORS

by

© S.J. Harrison

A thesis
presented to the
Faculty of Graduate Studies and Research
in partial fulfillment of the requirements
for the degree of
Master of Engineering

Carleton University
Ottawa, Canada
September, 1982
The undersigned recommend to the Faculty of Graduate Studies and Research the acceptance of the thesis:

Meteorological Effects on Thermal Performance Tests of Flat-Plate Solar Collectors

Submitted by, S.J. Harrison, B.Eng., in partial fulfillment of the requirements for the degree of Master of Engineering

Carleton University
October 15, 1982
ABSTRACT

A theoretical analysis of the effects of meteorological factors on the thermal performance of flat-plate solar collectors has been conducted. Theoretical results were determined by computer analysis of the optical and thermal performance of a typical flat-plate solar collector. Results are presented for the effects of meteorological and operational factors on component heat transfer and the combined effect on plots of Collector Thermal Efficiency.

An experimental determination of solar collector thermal performance was conducted for a variety of meteorological conditions. The results of the theoretical analysis and experimental data compare favorably. The feasibility of normalizing experimental data to standard meteorological conditions by theoretical analysis was investigated.

Results indicate that standard representations of solar collector thermal performance are not adequate to account for changing meteorological conditions.
ACKNOWLEDGEMENT

The author wishes to acknowledge the support and encouragement of the Division of Building Research and the Solar Energy Project Office of the National Research Council of Canada, and to express his appreciation to the people who assisted in this project. These include:

- J.R. Sasaki of NRCC for his encouragement during the project,
- Prof. J.T. Rogers of Carleton University for his guidance and patience during the project,
- L.P. Chabot for his assistance in setting up and conducting the experimental portion of the project,
- M. Bernier for assisting in reduction of the large amount of data collected,
- D. Scott of DBR for preparing the graphics for the final report,
- S. Nussbaumen and B. Gallimore for typing the report, and
- L.L. Hamelin for providing the computer program to calculate fluid properties.

This project would not have been possible without the cooperation of Carleton University and the National Research Council of Canada, and the author wishes to thank both parties. Portions of this research were funded under NSERC Grant A6342.
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NOMENCLATURE

A  area of collector surface
A_c collector reference area, i.e., aperture or gross collector area

C_p specific heat
C_b bond conductance

D  diameter of tubes
D_h hydraulic diameter
D_1 inside diameter

F  fin efficiency factor
F' collector efficiency factor

F_R collector heat removal factor

G  transfer fluid mass flow per unit collector area

h  heat transfer coefficient

I  total solar radiation measured in the collector plane

k  thermal conductivity

K  length extinction coefficient

K_1 parameter eqn.
K_2 parameter eqn.

L  collector component length, thickness

m mass flow rate for collector

n index of refraction

n_t number of flow tubes

Nu  Nusselt Number
$W_{per}$ wetted perimeter around fluid passageway

$Pr$ Prandtl Number, $\equiv C_p \mu / k$

$p$ pressure

$Q$ rate of energy transfer

$Q_u$ useful energy collected

$q$ heat transfer rate

$Ra$ Rayleigh Number

$R_A$ sky atmospheric longwave radiation on a horizontal surface

$R_A(s)$ sky atmosphere longwave radiation incident on an inclined surface

$Re$ Reynolds Number based on $D_h$

$R_G$ ground source longwave radiation

$R_G(s)$ ground source longwave radiation incident on an inclined surface

$R(s)$ total longwave radiation incident on an inclined surface

$S$ collector slope

$T$ temperature

$U$ thermal conductance

$V_w$ wind speed

$X$ parameter eqn. 5.

**Greek**

$\alpha$ absorptance

$\varepsilon$ emittance

$n$ thermal efficiency based on reference area
θ  angle between surface normal and beam radiation
θ_2  refraction angle
λ  wavelength
μ  absolute viscosity
ν  Kinematic viscosity
ρ  reflectance
σ  Stefan-Boltzmann Constant, (5.67x10^-8 W/m²K^4)
τ  transmittance
(τα)_e  effective transmittance absorptance product of the glazing and absorber plate

Subscripts
a  air, ambient
b  back side
c  cover, by convection
c-a  cover-to-ambient
d  diffuse
d_s  diffuse from sky
d_g  diffuse from ground
d_p  dew point
e  effective, edge
f  heat transfer fluid
f_1, f_0  inlet fluid, outlet fluid respectively
g  ground
i  insulation
L  overall loss
LW  Longwave radiation
o  fluid-to-ambient loss
p  absorber plate
p-c  plate-to-cover
p-s  absorber plate to sky
pm  mean for absorber plate
m  mean
r  by radiation
s  sky
st  stationary
T  top
W  wind induced
α  considering absorptance only
II,1  two components of polarization, parallel and perpendicular
1. INTRODUCTION

The increase in the use of solar energy for space and service-water heating has resulted in the introduction of a number of solar heating components, notably collectors, to the Canadian market. This has increased the need for solar component performance standards that will provide the users of solar heating systems with some assurance of functional performance, reliability and durability.

One component of these requirements is thermal performance testing. Collector thermal performance data is used (1) by purchasers and certifying agencies to compare the relative worth of competing products (i.e., ranking) and (2) by designers to predict the thermal performance of a system utilizing the evaluated collector. The thermal performance of a solar collector may be derived by analysis (1,2) using the known heat transfer and optical properties of the constituent materials or by experimental means. As many proprietary collectors are fabricated with materials and methods that are inadequately described in terms of their optical and heat transfer properties, it has become common practice to determine the performance characteristic of collectors by testing them with natural or simulated solar radiation.

The representation of thermal performance of flat plate solar collectors which is widely used is of the form originally derived by Hottel and Whillier (2) and later modified by Bliss (3). It is a simplified model of the thermal performance as a number of parameters are not accounted for.
These factors are often thought to represent only second-order effects but as the requirements for accuracy and repeatability increase, their influence becomes more significant. Recently, a test procedure for rating the thermal performance of solar collectors has been developed (4), based on the Hottel, Whillier, Bliss, model and has been widely used throughout North America and Europe. To assess the validity of this test procedure, a number of round-robin test programs have been performed with alarming results (5,6,7). Tests performed at different laboratories on similar test samples produced largely different results. Much of the variation was attributed to instrument error but an analysis of the meteorological conditions encountered during the tests indicated that they influenced the results. Other studies (8,9) have also indicated that environmental factors cause variations in the performance of solar collectors and recently Lumsdaine (10) and Bruck et al (11) have shown their effect on thermal performance test results.

In the light of these findings and realizing the requirements for repeatable and accurate performance ratings, this research was undertaken to determine the effect of meteorological conditions on thermal performance test results for flat plate solar collectors.

It should be noted that it has been shown (10,12) that factors other than those related to environmental conditions, (e.g., collector orientation, heat transfer fluid flow rate and physical properties, etc.), do affect the thermal performance of solar collectors. These factors, though, were considered outside of the scope of this study and as such were only considered in regard to the theoretical and experimental analysis of meteorological influences on collector thermal performance test results.
FIGURE 1
TYPICAL FLAT PLATE SOLAR COLLECTOR
2. THERMAL PERFORMANCE OF SOLAR COLLECTORS

2.1 Energy Balance for Flat-Plate Solar Collectors

A flat-plate solar collector, (figure 1) usually consists of an insulated box enclosing an absorbing surface with a transparent cover on the face to allow the entry of solar radiation. A heat transfer medium, (e.g., air, water, or glycol etc.), is circulated through flow channels attached to or part of the absorber plate.

The thermal performance of a solar collector may be described in terms of an energy balance that relates the solar energy incident on the collector to the energy removed by the heat transfer fluid and the thermal losses from the collector enclosure. This is given by:

\[ A_c (I (\tau \alpha)_{e}) = Q_u + Q_L + Q_s \]  

(1)

where

- I = rate of incident solar radiation on the solar collector per unit area
- \((\tau \alpha)_{e}\) = transmittance-absorptance product of the cover
- \(A_c\) = area of the collector aperture
- \(Q_u\) = rate of energy extraction from the collector by the heat transfer fluid
- \(Q_L\) = rate of energy loss from the collector to the surroundings
- \(Q_s\) = rate of energy storage in the collector

It is important to note that implicit in this simple heat balance is the fact that each of these terms is the sum of complex heat transfer
mechanisms including conduction, convection, and radiation heat transfer.

2.2 The Hottel, Whillier, Bliss Model of Thermal Performance

The theoretical performance of flat-plate solar collectors is described for steady-state conditions (2), where the rate of energy extracted from the collector is equal to the rate of radiative energy absorbed, minus the rate of heat loss to the surroundings, \(Q_s = 0\).

This is expressed as:

\[
\frac{Q_u}{A_c} = I(\tau_a) - U_L (T_p - T_a)
\]  
(2)

where:

- \(U_L\) = heat loss coefficient for the collector, \(W/m^2 K\)
- \(T_p\) = average temperature of the collector absorber surface, °C
- \(T_a\) = ambient air temperature, °C

For the purpose of system design, it is convenient to relate the performance of the solar collectors to the temperature of the heat transfer fluid, as the plate temperature is usually not known. Bliss (3) derived two collector efficiency factors, \(F'\) and \(P_R\), to allow the use of either the mean or inlet fluid temperature in the collector equation. These are given by:

\[
\frac{Q_u}{A_c} = F' \left[ I(\tau_a) - U_L (T_m - T_a) \right]
\]  
(3)

\[
\frac{Q_u}{A_c} = P_R \left[ I(\tau_a) - U_L (T_{fi} - T_a) \right]
\]  
(4)
where \( F' \) = collector efficiency factor

\[ F_R = \text{collector heat removal factor} \]

\[ T_m = \text{average temperature of fluid in the collector °C (i.e., arithmetic mean of inlet and outlet temperature)} \]

\[ T_{fi} = \text{inlet temperature of the fluid °C} \]

The two relationships are correlated according to reference (3) by:

\[ F_R = F' \left( \frac{1 - \exp(-X)}{X} \right) \]  \( (5) \)

where; \( X = F' U_L \left( G \ C_p \right)^{-1} \)

\[ G = \text{mass flow rate of the fluid per unit area of collector,} \]

\[ \text{(kg/s)/m}^2 \]

\[ C_p = \text{specific heat of transfer fluid J/(kg.K)} \]

The ratio of collected useful energy to solar energy incident on the collector is the collection efficiency, \( \eta \):

\[ \eta = \frac{Q_u}{I_{Ac}} = F_R \left( \tau a \right) e - F_R U_L \left( \frac{T_{fi} - T_a}{I} \right) \]  \( (6) \)

If values of \( \eta \) are plotted versus corresponding values of \( (T_{fi} - T_a)/I, a \) straight line with an intercept \( F_R \left( \tau a \right) e \) and slope \( F_R U_L \) will result, figure 2. Values of \( F_R \left( \tau a \right) e \) are good indicators of optical efficiency and values of \( F_R U_L \) are representative of the heat loss from the collector.
FIGURE 2

IDEALIZED SOLAR COLLECTOR PERFORMANCE CURVE

PLOT OF SOLAR COLLECTOR EFFICIENCY

TEST PARAMETERS
HEAT TRANSFER FLUID ........ WATER
MASS FLOW RATE ................ 0.044 kg/s
GROSS COLLECTOR AREA ........ 2.26 m²

COLLECTOR NO. 5/1981
TESTING DATE MARCH 30, 1981

FIGURE 3

TYPICAL MEASURED COLLECTOR PERFORMANCE CURVE FOR FLAT-PLATE SOLAR COLLECTOR
2.3 Thermal Performance Ratings

Thermal performance ratings are used by solar system designers and purchasers of solar heating components to compare the performance and cost effectiveness of competing products.

Thermal performance results are utilized to produce "sticker" performance ratings (20). These usually take the form of values of average daily output for specific inlet temperatures. Values are directly derived by multiplying the solar collector thermal efficiency by the incident solar radiation for a standard day. Values of thermal efficiency are determined from the thermal performance test result at a specific inlet fluid temperature, ambient air temperature and hourly average values of solar radiation. Purchasers utilizing this type of rating refer directly to the energy collected on the standard day to compare competing products. The energy output at the required fluid temperature is combined with unit cost to arrive at the most cost effective product.

For large commercial installations, computer simulation programs are utilized to determine collector array output and subsequent system performance. These simulation programs input the solar collector characteristics, i.e. \( F_{R}(\tau) \) and \( F_{R} U \). Solar collector thermal efficiency is determined for hourly data, of solar radiation, ambient air temperature and fluid inlet temperature (1).

Simplified design methods exist which utilize long term average solar radiation and air temperature data for particular locations (73).
All of these procedures assume that a specific thermal performance test result fully characterizes a solar collector's performance, (some simulations include the effect of solar radiation incident angle), but most do not account for the effects of varying meteorological and operational parameters on solar collector thermal performance.

Finally, direct comparisons of the thermal performance shown on a performance curve are often made. In many instances the choice of product used has been determined on the basis of plus or minus a couple of efficiency points.

Contracts to supply solar heating equipment are often awarded on the basis of thermal performance test results. This has prompted solar collector manufacturers to demand a high degree of accuracy in performance ratings.

It is clear that users of solar collector test data should be aware of the factors that affect the test result and their uncertainties.
3. THERMAL PERFORMANCE TEST METHODS

3.1 Review of Test Methods and Procedures

A comprehensive review of existing thermal performance test methods was performed during 1974 by The National Bureau of Standards (8). From this, a formalized collector testing method was recommended including the testing procedure and facility to be used, and the required accuracy of the measurement instruments (13). A second test procedure, ASHRAE Standard 93-77, was introduced by the American Society of Heating, Refrigeration and Air-Conditioning Engineers in 1977 (4). Basically, the ASHRAE standard outlines the same efficiency tests as the NBS procedure, but includes additional tests for determining the transient response of the collector and the variation in performance due to increasing incident angle. (The angle between the direct solar beam and an outward drawn normal to the plane of the collector aperture).

These procedures call for steady-state tests to be performed over a range of collector inlet fluid temperatures under clear sky and steady radiation conditions at times near solar noon. Values of experimental efficiency, "\( \eta \)" are determined by measuring, simultaneously, the mass flow rate, inlet and outlet fluid temperatures and the solar insolation. The instantaneous efficiency is determined by the ratio of energy collected by the heat transfer fluid to the solar energy incident on the collector. This is given by:

\[
\eta = \frac{Q_u}{IA_c} = m \frac{C_p}{IA_c} \left( T_{fo} - T_i \right)
\]  

(7)
where; \( m \) - the mass flow rate of heat transfer fluid through the collector, kg/s

\( C_p \) - the specific heat of the heat transfer fluid, J/kg°C

\( T_{fo} \) - outlet temperature of the fluid, °C

Experimentally derived values of collection efficiency may then be plotted against the appropriate parameter, i.e., \((T_{fi} - T_{ra})/I\), and the values of \(F_{eR}(ra)\) and \(F_{L}(U_1)\) determined by the intercept and the slope of the performance curve, figure 3. A complete description of the ASHRAE test procedure and apparatus is given in references (4) and (8).

3.2 Problems Related to Thermal Performance Testing

The performance testing of solar collectors is a relatively complex procedure. ASHRAE Standard 93-77 (4) was intended to provide a standardized test procedure to enable the characterization of the thermal performance of solar collectors in a universal plot. These plots could then be compared in order to rate products and predict solar system performances. This testing procedure and performance representation is based on the simplified Hottel, Whiller, Bliss model of thermal performance described previously. In this representation a number of parameters are not accounted for, such as wind speed and direction, absolute ambient air temperature, ground reflectance, collector tilt, percentage of diffuse radiation, and the effects of the intensity of solar radiation on collector thermal losses. These factors have in the past been considered to account for only second order effects and it was thought satisfactory just to limit the extreme values during data selection. The test procedure sets ranges of acceptable values for
windspeed, solar radiation, and ground reflectance. The ambient air
temperature is allowed to vary only 30°C during a single test and the data
is limited to clear sky periods when the diffuse solar radiation level is
low and the collector performance is at steady state.

This presents two problems; 1. a large number of relatively clear
sky periods are required to yield a full set of data and at certain times of
the year this may require a long test period during which the variability of
the weather conditions can cause a high degree of scatter in the data making
interpretation difficult. 2. The requirement for limited environmental
conditions during testing further limits the available testing time and
causes the time required for a full performance evaluation to be
lengthened.

The ASHRAE standard attempts to compromise between available test
periods and variability in the test results. The procedure in its existing
form would be suitable if all the tests were performed in a climatic region
where atmospheric variations from season to season were limited and clear
sunny days were abundant. Such is the case for the south western United
States where in fact one of the largest test facilities exists. Even at the
best locations, variations in atmospheric conditions do occur and cause
variability in the test results. The problem is much more severe in the
more northern regions of the U.S. and Canada where large seasonal variations
occur.

This effect was illustrated in a program organized by the National
Bureau of Standards intended to assess the validity of the test procedure.
In this "Round-Robin Test Program", thermal performance tests were performed
on two types of solar collectors by 21 test laboratories across the U.S. (5). The results, (figure 4), showed a large spread in the measured values of collector efficiency. The variation was attributed to systematic errors and the different environmental conditions experienced from facility to facility. A more recent "Round-Robin" test, performed under the organization of the International Energy Agency, (IEA) showed similar results but the variation that occurred was largely attributed to environmental factors experienced during the test and systematic errors (6).

These test programs indicate the problem with current test procedures and performance representations. Consistent results cannot be achieved when virtually identical test samples are tested to a similar procedure by different test laboratories. The variations that occurred were certainly due to systematic differences between test laboratories, random measurement uncertainty, sample variation and changing meteorological conditions, but it is not clear to what extent each factor has influenced the results. Theoretical analysis indicates that while random measurement uncertainty and sample variability cannot alone account for the variation in the test results, meteorological conditions could account for the variation as could systematic errors (7).

Lumsdaine (10) has also indicated possible sources of error in the testing of solar collectors. In particular he cites collector tilt angle, wind velocity, and ratio of direct to diffuse solar radiation as factors causing variations in test results. Lumsdaine suggests that test results are a function of a number of factors and that a modified test procedure
TEST RESULTS FROM DIFFERENT LABORATORIES FOR IDENTICAL SINGLE GLAZED FLAT PLATE SOLAR COLLECTORS

\[ \frac{(T_{fl} - T_a)}{I}, °C/(W/m^2) \]

FIGURE 4
RESULTS OF "ROUND-ROBIN TEST PROGRAM", CONDUCTED BY NATIONAL BUREAU OF STANDARDS
should be used where the effects of atmospheric conditions are further controlled. The ASHRAE test procedure implicitly assumes that the value of collector efficiency $\eta$ is only a function of $(T_{fi} - T_a)/I$ and that $F_R(\tau_e)$ and $F_{UL}$ are effectively constant. Bowen (14) and Shewen and Hollands (15) have elaborated on the problems of utilizing a constant value for the heat loss coefficient of solar collectors. Not accounting for environmental and operational conditions, may result in an under- or over-prediction of collector performance. Shewen and Hollands (16) have stated that if parameters $F_{UL}$ and $F_R(\tau_e)$ were truly constant, the standard test procedures presented in ASHRAE Standard 93-77 would constitute a complete thermal performance test on the collector. However, many parameters are temperature dependent (particularly $F_{UL}$) and result in a convex-up plot rather than a straight line when $\eta$ is plotted versus $(T_{fi} - T_a)/I$ as shown in figure 3. As a result the ASHRAE plot is not a universal plot for a given collector, but dependent on the values of the meteorological conditions experienced during the test. Variations of these quantities over a test period can cause scatter in the data and variation between specific tests can lead to biased results. This makes comparisons between collector types and accurate performance predictions impossible.

To overcome this difficulty it has been argued (7,16) that test data should be referred to conditions of standardized values of $T_a$ and $I$. If data were not measured under these standard conditions, an agreed procedure should be used to correct the values to standard conditions. It is proposed (16) that these standardized values should be representative of average Canadian conditions, with a weighting toward the winter if the collectors
are to be used for solar heating systems. The following standard values
were suggested (16);

\[ T_a = 0^\circ C \]

\[ I = 630 \text{ W/m}^2 \]

Streed (5) attempted to normalize round-robin test results to
standard conditions but the results were confounded by both systematic
measurement errors between the participating test facilities and uncertainty
in the uniformity of optical and thermal properties of the test sample.
Bruck (11) has shown the effects of the various atmospheric conditions
experienced during a test by computer analysis but did not verify the
results by experimental means.

While normalization of test results to standard test conditions has
been suggested as one approach to direct comparison, another possibility is
to modify the basic Hottel, Willier, Bliss model to account for more of the
environmental parameters experienced during a test. Modifications of this
type have been suggested by Shewen and Hollands (15) and Tabor (17,18).

3.3 Problem Definition and Approach

As outlined in the previous section, the ASHRAE representation of
thermal performance does not produce a universal plot for a given solar
collector. Data indicates that other factors not accounted for in the
Hottel, Willier, Bliss Model affect the test results making detailed
comparisons impossible. Studies have shown that variations in test results
do occur (5,6), although the influence of any one factor has not been demonstrated. This lack of definitive results was largely due to the possibility of a number of factors influencing the result of previous studies. Possible systematic errors or sample variability introduced major uncertainties into the results of the Round-Robin test programs undertaken in the past. A separate study to determine the effect of meteorological factors on solar collector test results is required. This study should be conducted in a manner that eliminates the effects of sample variability and systematic measurement error. These objectives may be met by two methods;

1. A theoretical analysis of the basic heat transfer properties associated with solar collector thermal performance is performed to determine the relative effects of each of the atmospheric parameters on the thermal performance of solar collectors. The computer analysis may be performed in a systematic manner by varying each parameter separately to determine its effect.

2. The performance of a single solar collector test sample is measured under a variety of atmospheric conditions, utilizing a single test facility for all tests. The use of a single test sample and test facility ensure that systematic errors and sample variability are eliminated when the relative effects of atmospheric conditions on collector thermal performance are evaluated.

The project described herein consists of the experimental testing of a solar collector to determine its thermal performance under a variety of atmospheric conditions spanning the range of values experienced in the Canadian climate. Values of wind velocity, solar radiation intensity and
percent diffuse component, absolute ambient air temperature, and heat transfer fluid temperature were measured for each operational condition. A single glazed flat plate solar collector (see Appendix A) was used for the evaluation as it is described in many studies (5,6,7) and its optical and thermal properties are well known.

Data was recorded for a period of time spanning almost a full calendar year ensuring that a range of atmospheric conditions were experienced. The data was recorded on magnetic tape for subsequent analysis by computer. Only data that met the basic requirements of the ASHRAE test standard (i.e., steady state operation, minimum solar radiation, etc.), were used for the analysis.

A major phase of this project consisted of the writing of a computer program to simulate the thermal performance of the solar collector. The computer program utilizes fundamental heat transfer relationships to simulate the collector's performance at specific atmospheric conditions. The computer model allows the systematic analysis of each of the atmospheric parameters to determine its effect on collector performance. Once the effects of these parameters is determined the analysis may be used to evaluate the feasibility of normalizing the data taken during the experimental portion of this project to standard conditions. The effectiveness of the theoretical analysis in normalizing the experimental data to standard conditions is determined by the agreement of the generated data with the experimental results and the reduction of scatter in the experimental data.
4. HEAT TRANSFER ANALYSIS

4.1 Heat Transfer in Solar Collectors

The theoretical equation for the thermal performance of flat plate solar collectors is given in section 2 as:

\[ Q_u = A_c \left( I(\tau_a)e \right) - Q_L - Q_s \]  

(8)

Assuming \( Q_s = 0 \), then, the rate of energy extracted from the collector is the difference between the rate of solar energy absorbed by the absorber plate, \( I(\tau_a)e \), and the rate of thermal energy loss to the surroundings, \( Q_L \), where \( Q_L = U_L A_c (T_p - T_a) \).

Optical and thermal loss mechanisms are shown diagramatically in figure (5) for a conventional flat plate solar collector. Optical losses

![Diagram](image-url)
include absorption and reflection of the incident solar beam at the glazing of the collector and reflection at the absorber surface. Thermal losses from the collector consist of conduction losses through the back and edge of the collector to the surrounding environment, and conduction, convection and radiation losses through the front of the collector.

In the simplified representation of thermal performance, the values of \( (\tau_o) \) and \( U_L \) are considered to be constant for all atmospheric conditions. This is, in fact, not true as both the optical and heat transfer properties of the solar collector are effected by meteorological conditions.

The energy balance on a single glazed flat plate solar collector may be represented in simple terms by the thermal network shown in figure 6.

In actual fact, the resistance networks for the heat transfer in a solar collector are quite complex. To assess the effects of atmospheric factors, we may first investigate the factors that affect the overall heat loss coefficient, \( U_L \) and separately those that affect the optical properties, \( (\tau_o) \). To ultimately assess the overall effect on the thermal performance of a solar collector, the interaction of these two effects must be evaluated.
The following sections will present the analysis of the steady state thermal performance of the Chamberlain flat plate solar collector, described in Appendix A. This collector is of conventional design and construction and is the model used for studies at NRCC. The analysis is derived using conventional heat transfer practice and follows closely the relationships presented by Duffie and Beckman (1, 40) and Kreith and Kreider (19). The general analysis, while specifically derived for the Chamberlain collector, will be applicable to other collectors of similar design. Transient effects, while present in the operation of real systems, are not allowed during thermal performance testing and so will not be considered in this analysis.

4.2 Heat Loss Coefficient \( U_L \)

If the following assumptions are applied, the heat loss network for a solar collector may be represented as shown in figure 7.
1. The solar collector is thermally in steady state.

2. The temperature distribution from the top to the bottom of the absorber plate is uniform.

3. Heat flow is one-dimensional through the covers and back insulation.

4. The headers connecting the absorber plate tubes represent only a small percentage of the absorber plate.

5. The sky can be represented as a black-body sink for infrared radiation at an equivalent sky temperature.

6. The irradiation of the collector plate is uniform.

\[
\begin{align*}
\text{FIGURE 7} \\
\text{THERMAL NETWORK FOR HEAT LOSS FROM THE ABSORBER PLATE SHOWING EDGE, BACK AND TOP LOSSES}
\end{align*}
\]

The heat loss from the absorber plate to the surrounding atmosphere may be refined as shown in figure (7) to show the back and edge heat losses.
through the collector casing. For most solar collectors the relative values of these two terms is small compared with the losses through the collector front glass face. The rate of heat loss through the edge and back of the solar collector is relatively insensitive to variations in atmospheric conditions other than those of the temperature difference between $T_p$ and $T_a$.

The value of $Q_L$ is the rate of energy loss from the collector to the surroundings. These losses will be made up of losses through the back and edges of the collector and losses through the top of the collector;

$$Q_L = q_{\text{back}} + q_{\text{edge}} + q_{\text{top}} \quad (9)$$

Relating each of these losses to the temperature difference between the mean absorber plate temperature, $T_{pm}$, and the ambient air temperature, $T_a$, the expression becomes:

$$Q_L = A_{\text{back}} U_{\text{back}} (T_{pm} - T_a) + A_{\text{edge}} U_{\text{edge}} (T_{pm} - T_a) + A_{\text{top}} U_{\text{top}} (T_{pm} - T_a) \quad (10)$$

If we further relate each of these values to the reference area of the solar collector, the heat loss per unit collector reference area, $A_c$, is;

$$\frac{Q_L}{A_c} = \left[ \frac{A_{\text{back}} U_{\text{back}}}{A_c} + \frac{A_{\text{edge}} U_{\text{edge}}}{A_c} + \frac{A_{\text{top}} U_{\text{top}}}{A_c} \right] (T_{pm} - T_a) \quad (11)$$
If we let
\[
U_b = \frac{A_{\text{back}}}{A_c} U_{\text{back}} \quad ; \quad U_T = \frac{A_{\text{top}}}{A_c} U_{\text{top}} \quad ; \quad U_e = \frac{A_{\text{edge}}}{A_c} U_{\text{edge}}
\]
(12)
then
\[
\frac{Q_L}{A_c} = \left[ U_b + U_e + U_T \right] (T_{\text{pm}} - T_a)
\]
(13)

The value of \( U_b + U_e + U_T \) is usually given the name Heat Loss Coefficient, \( U_L \),

i.e.,
\[
U_b + U_e + U_T = U_L
\]
(14)

Thus, the relationship for heat loss can then be given as
\[
Q_L = \frac{A_{\text{c,L}}}{A_c} (T_{\text{pm}} - T_a)
\]
(15)

4.2.1 Back Heat Loss Coefficient

The energy loss through the back of the collector is determined by the thermal resistance of the back insulation and the collector housing and film coefficients for the outer surface. In most cases, (and the one being considered), the value of heat loss may be accurately predicted using the resistance of the insulation only since the thermal resistance of the insulation constitutes the governing resistance to heat flow from the back.
The effect is even further reduced as edge and back losses are usually a small portion of the total loss from the collector. See results in Appendix C. The back heat loss coefficient is then given by:

\[ U_b = \frac{U_{back} A_{back}}{A_c} \]

where

\[ U_{back} = \frac{k_i}{L_i} \]

giving

\[ U_b = \frac{k_i A_{back}}{L_i A_c} \]

4.2.2 Edge Heat Loss Coefficient

Accurate derivations of the edge losses of solar collectors is quite complicated. Relationships for determining the average values of edge loss are given in references (19) and (22) for specific configurations. The edge Heat loss coefficient is given as:

\[ U_e = \frac{k_e A_{edge}}{L_e A_c} \] \( (17) \)

where \( k_e \) is the thermal conductivity of the edge insulation,

\( L_e \) is the thickness of the edge insulation.

Edge losses often vary from collector make to make depending on the details of the design. Consequently combined values of \( U_{back} \) and \( U_e \) are often determined experimentally.
4.2.3 Top Heat Loss Coefficient

Heat loss through the top of the solar collector is by both convection and radiation heat transfer. Heat is transferred from the absorber plate to the cover plate by convection across an inclined air layer and also by radiative exchange between the plate and the cover glass. Some long-wave radiation is transmitted through the glass from the absorber plate to the atmosphere but since glass is almost totally opaque to long-wave radiation, this accounts for only a small amount of the heat loss. This is not always the case for some thin films and plastics, as they may be effectively transparent to long-wave infrared radiation.

The energy transferred to the cover from the absorber plate is lost to the atmosphere by convection to the ambient air (often enhanced by wind), and by radiation exchange from the cover plate to the sky and the surroundings. The radiation leaving the cover plate is assumed to be radiating to a black-body source for infrared radiation at an effective sky temperature. Relationships for effective sky temperature, $T_s$, are discussed in section 4.4.2.

A detailed heat transfer network for the top-loss heat transfer coefficient is given in figure 8, showing both the radiative and convective heat loss paths.
FIGURE 8
THERMAL NETWORK FOR HEAT-LOSS FROM THE TOP
OF A SOLAR COLLECTOR SHOWING CONVECTIVE
AND RADIATIVE COMPONENTS

The simplified performance representation described in section 2.0
assumes that $U_L$ (and $U_{top}$) are constant. It may be seen from figure 8 that,
in fact, $U_{top}$ is a function of both $T_a$, $T_p$ and $T_{sky}$ and the values of $h_{w,c-a}$,
$h_{r,c-s}$, $h_{c,p-c}$ and $h_{r,p-c}$. These terms are a function of collector
orientation and wind velocity. Gross assumptions that are commonly made are
that $h_w$ is constant regardless of wind speed and that $T_s$ is equal to the
ambient air temperature $T_a$. The validity of these assumptions will be
described in detail in the subsequent sections.

The relationship for heat transfer from the collector absorber plate
to the coverglass is given by:
\[ q_{p-c} = q_{\text{rad},p-c} + q_{\text{conv},p-c} \]  

The value for the radiative exchange between the plate and cover can be approximated by the relationship for radiative exchange between infinite parallel flat plates (4).

\[ q_{\text{rad},p-c} = \frac{A_{\text{top}} \sigma (T_{\text{pm}}^4 - T_{\text{c}}^4)}{(1/\varepsilon_p) + (1/\varepsilon_c) - 1} \]  

This relationship should be valid for almost all flat plate collectors as the distance between the cover and absorber plate is small relative to the area of the plates. Subsequently edge effects should be minimal.

The heat transfer due to convection is given by:

\[ q_{\text{conv},p-c} = A_{\text{top}} c_{\text{p-c}} (T_{\text{pm}} - T_{\text{c}}) \]  

Converting the relationship for \( q_{r,p-c} \) we may derive the radiation heat transfer coefficient \( h_{r,p-c} \):

\[ h_{r,p-c} = \frac{q_{\text{rad},p-c}}{A_{\text{top}} (T_{\text{pm}} - T_{\text{c}})} \]
thus

\[
h_{r,p-c} = \frac{\sigma (\frac{T_{pm}^4}{T_c^4} - T_c^4)}{[(1/\varepsilon_p)^{-1} + (1/\varepsilon_c)^{-1}]T_{pm} - T_c}
\]

which can be reduced to

\[
h_{r,p-c} = \frac{\sigma (T_{pm}^{2+T_c^2} + (T_{pm}^{2+T_c^2})}{(1/\varepsilon_p)^{-1} + (1/\varepsilon_c)^{-1} - 1}
\]

The value of the convective heat transfer coefficient, \(h_{c,p-c}\)
is determined by the relationship;

\[
h_{c,p-c} = \text{Nu} \left( \frac{k_a}{L_{p-c}} \right)
\]

The relationships for the \(h_{c,p-c}\), \(\text{Nu}\), the Nusselt number and \(k_a\), the thermal conductivity of air are presented in sections 4.2.3.3. The heat transfer from the absorber plate to the cover can now be expressed as;

\[
q_{p-c} = A_{top} U_{p-c} (T_{pm} - T_c)
\]

where

\[
U_{p-c} = h_{c,p-c} + h_{r,p-c}
\]
It should be noted that the heat loss from the absorber plate to the cover glass is the same value as the heat transferred from the cover glass to the atmosphere.

\[ q_{p-c} = q_{c-a} \]  \hspace{1cm} (27)

The heat loss from the cover to the atmosphere is by convection and by radiation to the sky;

\[ q_{c-a} = q_{\text{rad, c-s}} + q_{\text{conv, c-a}} \]  \hspace{1cm} (28)

where

\[ q_{\text{conv, c-a}} = A_{\text{top}} h_w (T_c - T_a) \]  \hspace{1cm} (29)

where \( h_w \) is the wind induced heat transfer coefficient described in section 4.2.3.2.

The radiation loss from the cover to the sky is given by;

\[ q_{\text{rad, c-s}} = A_{\text{top}} \varepsilon_c \sigma (T_c^4 - T_s^4) \]  \hspace{1cm} (30)

as expressed as radiation heat transfer coefficient

\[ h_{r,c-a} = \frac{\varepsilon_c \sigma (T_c^4 - T_s^4)}{(T_c - T_a)} \]  \hspace{1cm} (31)

where \( T_s \) is given by the expression developed in section 4.2.3.2. The resultant heat loss is given by
\[ q_{c-a} = A_{top} \left( h_w + h_{r,c-s} \right) \left( T_{c} - T_{a} \right) \]  

\[ = A_{top} \left( U_{c-a} \right) \left( T_{c} - T_{a} \right) \]

where:

\[ U_{c-a} = h_w + h_{r,c-s} \]  

Heat transfer by radiative exchange between the absorber plate and the sky is given by;

\[ q_{rad,p-s} = A_{top} \tau_{LW} e \sigma \left( T_{pm}^4 - T_{s}^4 \right) \]

where \( \tau_{LW} \) is the transmittance of the cover to long-wave radiation.

This radiation heat transfer coefficient is given as:

\[ h_{r,p-s} = \frac{\tau_{LW} e \sigma \left( T_{pm}^4 - T_{s}^4 \right)}{(T_{pm}^4 - T_{a}^4)} \]  

The overall heat transfer coefficient for top heat loss may now be expressed as

\[ U_{top} = \left[ \frac{1}{U_{p-c}} + \frac{1}{U_{c-a}} \right]^{-1} + h_{r,p-s} \]  

The heat loss for the collector top per collector unit area is then:

\[ \frac{q_{top}}{A_c} = \frac{A_{top}}{A_c} U_{top} \left( T_{pm} - T_{a} \right) = U_{top} \left( T_{pm} - T_{a} \right) \]

for \( A_c = A_{top} \)
4.2.3.1 **Effective Sky Temperature**

To evaluate the radiation exchange between a surface, of area $A$, and the sky, the sky can be considered as a blackbody at an effective sky temperature $T_s$. In this case the net radiation exchange, $Q_{rad}$, between a horizontal flat plate and the sky is given by:

$$
\frac{Q_{rad}}{A} = \varepsilon_p \sigma (T_p^4 - T_s^4)
$$

(37)

In the simplified representation of collector thermal performance described in section 2.0 it is usually assumed that the effective sky temperature is equal to the ambient air temperature. Many researchers have shown that this assumption is often not true, (1, 21, 22, 23, 25,26).

The difference in sky and ambient temperature is a result of radiative exchange between the atmosphere and the radiating surface. Atmospheric radiation originates from constituents within the atmosphere. The intensity of the radiation emitted at any wavelength is dependent upon the partial pressure of the constituents, their temperature and their thickness. Water vapour and carbon dioxide are the principle constituents with respect to atmospheric radiation, the former being the most important. Both show absorptive (and emissive) characteristics at selected wavelengths over the entire spectrum. In the wavelength region 8.5 to 13.0 $\mu$m, atmospheric water vapour is largely transparent to longwave radiation, but at all other wavelengths it can be considered practically opaque to longwave radiation (24). Clouds are powerful sources of atmospheric radiation and in
general, the effect of clouds is to close the "atmospheric window" in the 8.5 to 13.0 µm region. Clouds will therefore always increase the amount of atmospheric radiation relative to the clear sky condition. This will result in higher effective sky temperatures as compared to the clear sky case where the net radiative exchange is to the much colder upper atmosphere. Solar collector thermal performance tests are only conducted during relatively clear sky conditions although the amount of moisture in the atmosphere may vary for different tests. In actual system operation large amounts of cloud cover may be present but these periods will usually constitute periods when the available solar radiation is low. During collector testing though, large differences in ambient air temperature and effective sky temperature may be present.

Several relationships have been proposed to relate $T_s$, for clear skyes, to other meteorological variables. This has been undertaken as effective sky temperature has not been routinely measured in the past. Relatively inexpensive instruments for direct measurements have recently been developed but their accuracy is still in question (21).

Swinbank (23), has related sky temperature to local air temperature by the relationship

$$T_s = 0.0552 \left( T_a \right)^{1.5}$$

(38)

where $T_s$ and the ambient air temperature, $T_a$ are both in degrees Kelvin. Bliss (24), has related effective sky temperature to the water content of
the air and its temperature. The relation uses dew point temperature, $T_{dp}$;

$$T_s = T_a \left[ 0.8 + \frac{T_{dp} - 273}{250} \right]$$  \hspace{1cm} (39)

Willier (22) recommends subtracting 6°C from the air temperature

$$T_s = T_a - 6°C$$  \hspace{1cm} (40)

although values significantly lower have been recorded (25).

Unsworth and Monteith (28), have recently derived a relationship for incoming atmospheric radiation, $R_A$, from the sky during clear sky conditions

$$R_A = p + q \cdot \sigma T_a^4$$  \hspace{1cm} (41)

where $p$ and $q$ were found to be $-119 \pm 16 \text{ W/m}^2$ and $1.06 \pm 0.04$ respectively from measurements in England. The relationship has an estimated uncertainty in a single estimate $R_A$ of $\pm 30 \text{ W/m}^2$, (or $\pm 6 \text{ K}$ at $T_a = 293 \text{ K}$).

The value of effective sky temperature may be simply determined, e.g.,

for the case $T_a = 20°C = 293 \text{ K}$

$$R_A = -119 + 1.06 \left( 5.67 \times 10^{-8} \right) (293)^4$$

$$= 324 \text{ W/m}^2$$
The effective sky temperature is determined by treating this radiation as equivalent to that coming from a black body radiator at a temperature equal to $T_b$.

$$T_s = \frac{324}{5.67 \times 10^{-8}} \frac{1}{\gamma}$$

$= 275 \, K = 2^\circ C$

Swinbank's relationship gives a value of $3.8^\circ C$. Figure (9) compares values derived by the two relationships (21).
It may be noted that the depression of the $T_s$ relative to $T_a$ is significant and subsequently should be considered in the heat transfer analysis. It should be noted that while this difference may be large, the solar collector is usually tilted at some angle to the horizontal and as such views the ground in front of it as well as the sky. The ground is usually at a temperature very close to the ambient air temperature. The net result is that the collector "sees" two surfaces; one at $T_s$ and one at $T_a$.

Values of longwave radiation incident upon inclined surfaces have been derived (26). The radiation incident upon a surface inclined at an angle to the horizontal, figure (10) is a combination of atmospheric and ground radiation, i.e.,

$$ R(s) = R_A(s) + R_C(s) \quad (42) $$

where: $R(S)$ - is the incident longwave radiation on the surface, W/m²

$R_A(S)$ - is the incident atmospheric radiation on the inclined surface, W/m²

$R_C(S)$ - is the longwave radiation incident on the inclined surface from the adjacent ground, buildings, etc., W/m².

Inclined surfaces receive radiation from the atmosphere, the ground and any surrounding buildings, in the ratio of their respective configuration factors. Although it is possible to predict the longwave radiation originating from the atmosphere, there is very little information available regarding the magnitude of the ground radiation (26). The ground radiates in accordance with the temperature attained by its surface and its spectral emissivity. The ground temperature is dependent upon the nature of the surface, the prevailing climatic conditions and the heat gains and losses during the previous day. It is generally assumed that the ground radiation
seen by an inclined surface is equal to that emitted by a black body at
ambient air temperature, but if the emissivity of the ground is known then
the value of the ground component, \( R_G(S) \), incident on a surface inclined at
a slope \( S \) to the horizontal may be estimated by;

\[
R_G(S) = \varepsilon_g \sigma T_g^4 \cdot \frac{(1-\cos S)}{2}
\]  
(43)

The value of \( R_A(S) \) is more complex as atmospheric radiation is not
isotropic and varies across the sky. Cole (26) has derived a relationship
for atmospheric radiation incident on an inclined surface, for clear skies,

\[
R_A(S) = R_A \cdot K_1 + K_3 \cdot b \sigma T_a^4
\]  
(44)

where; \( R_A \) - is calculated for the horizontal case as before
\( b \) - is a constant = 0.09
\( K_1 \) and \( K_2 \) - are given in table (1) as taken from reference (27).

Considering the previous example for a surface sloped at 60° to the
horizontal;

As calculated for the horizontal case \( R_A = 324 \text{ W/m}^2 \).

\[
R(60) = R_A(60) + R_G(60)
\]

and

\[
R_A(60) = R_A \cdot K_1 + K_3 \cdot 0.09 \cdot \sigma T_a^4
\]
FIGURE 10

SURFACE INCLINED AT SLOPE "S"

<table>
<thead>
<tr>
<th>Surface inclination</th>
<th>$K_1$</th>
<th>$K_3$</th>
</tr>
</thead>
<tbody>
<tr>
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<tr>
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</tr>
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</tr>
<tr>
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<td>0.3381</td>
</tr>
<tr>
<td>90</td>
<td>0.5000</td>
<td>0.3457</td>
</tr>
</tbody>
</table>
from table (1) for 60° inclination, \( K_1 = 0.7500 \), \( K_3 = 0.2803 \), and 
\[ T_a = 20^\circ C = 293^\circ K \]
therefore:
\[
R_A(60) = 324 \times (0.7500) + 0.2803 \times 0.09 \times 5.67 \times 10^{-8} \times (293)^4
\]
\[ = 253.5 \text{ W/m}^2 \]

The value of \( R_G(60) \) is given by:
\[
R_G(60) = \frac{\varepsilon_g}{\sigma g} T_a^4 \left( \frac{1 - \cos 60}{2} \right)
\]
assuming that the ground is rough concrete with a value of \( \varepsilon_g = 0.94 \), (29) and \( T_g = T_a \),
then \( R_G(60) = 0.94 \times 5.67 \times 10^{-8} \times (293)^4 \times \frac{1 - \cos 60}{2} \)
\[ = 98.2 \text{ W/m}^2 \]

The value \( R(60) \) is then:
\[
R(60) = R_A(60) + R_G(60)
\]
\[ = 253.5 + 98.2 \text{ W/m}^2 \]
\[ = 351.7 \text{ W/m}^2 \]

the value of \( T_s \) for this case is then,
\[
T_s = \frac{351.7 + 98.2}{5.67 \times 10^{-8}}
\]
\[ = 280.6 \text{ K} = 7.6^\circ C \]

We see that the value of \( T_s \) at 60° tilt is 5.6°K warmer than the horizontal case due to longwave radiation emitted from the ground and seen by the inclined surface. Theoretical values of longwave radiation incident on an
inclined surface area shown in figure (11) as calculated by the above relations using the TSKY program described in section 4.6, and assuming $\varepsilon_g = 0.9$.

![Graph showing effective sky temperature for inclined surfaces (clear sky)](image)

**FIGURE 11**
EFFECTIVE SKY TEMPERATURE FOR INCLINED SURFACES (CLEAR SKY)

4.2.3.2 Wind Induced Heat Transfer Coefficient

The effect of wind velocity on the thermal performance of flat plate solar collector has been investigated by many researchers (1, 8, 42, 44).
While it is clear that wind considerably influences collector thermal performance through variations in $h_w$ \(^{(44)}\), the wind induced surface heat transfer coefficient, the relationship between $h_w$ and atmospheric wind is an area of considerable controversy.

The flow of wind over a solar collector is highly complex in nature due to variations in wind direction and velocity that result in complex 3-dimensional flow patterns. Nearby structures, obstructions and surface roughness affect the atmospheric boundary layer. This introduces considerable uncertainty into the determination of the flow field around real solar collector installations.

Experiments conducted outdoors on real solar collectors are complicated by the transient nature of the wind and solar radiation. The relatively long response time of the solar collector, absorber plate and glazing make the collection of meaningful data difficult if not impossible. For this reason, many recent studies \((8,44)\) to relate wind effects to the thermal performance of solar collectors have relied on computer analysis utilizing empirical relationships derived for surface heat transfer coefficients determined in wind tunnel experiments \((32, 33, 37, 38, 39)\).

Analysts \((1,8,22,30)\) in the past have used the relationship for forced convection heat transfer suggested by Jürges in 1924, \((33)\), which is based on the work done by Nusselt and Jürges in 1922 \((32)\). The relationship as presented by Duffie and Beckman \((1)\), is:

$$h_w = 5.7 + 3.8V$$ \((45)\)

where $h_w$ is the wind induced heat transfer coefficient in W/m\(^2\)K, and $V$ is the wind velocity in m/sec.
The original description of this work (32) was presented in German and was introduced into American engineering practice by McAdams in his classical book, "Heat Transmission" (34), in 1933. The relationship was adopted by ASHRAE, (31) for calculating wind induced heat losses from buildings. The fact the original text was in German seems to have limited later investigators' understanding of the original experiment and led to considerable speculation over the last 60 years as to the work's applicability to solar collectors.

Criticisms of the work (33) are summarized below with corresponding references:

1. The data was originally obtained for air flow parallel to a vertical copper plate which is typically not the case found in solar collector installations where wind may pass over the collector at an arbitrary angle. (36, 38, 39, 46).

2. The relationship does not account for the dimensions of the plate used for the experiment although classical heat transfer theory indicates that average heat transfer coefficient depends on the stream-wise length of the plate. Therefore results are only applicable to plates of the same dimension as used in the original experiment (38, 40).

3. The characteristics of the experiment are not defined, in relation to Reynolds number and turbulence intensity of the flow. The exact specification of V is not given (37, 38, 46).

4. Radiation heat transfer was probably not accounted for in the experiment (36, 38, 40).

The work of Nusselt and Jürges (32), has recently been translated into English by Serré, 1979, (35). A review of this translation indicates that the original work was performed with a 0.5 m square copper plate mounted vertically on one side of a wind tunnel. The flow over the plate
was uniform and radiation heat transfer was accounted for, negating criticism no. 4. It is interesting to note that authors state that the relation in wide scale usage (33) is only applicable to velocities below 5 m/sec. A relationship for wind speed above this value is presented in the original work. Reviewing the work of Nusselt and Jürges, Gladstone (47), points out that the turbulence intensity encountered during the tests was about 1.3%, whereas atmospheric wind has values ranged from 10 to 30%.

It may be concluded from the above discussion, that there is good agreement that the relationship widely in use (33), is inappropriate for application to solar collectors.

This seems to be the extent of the agreement of subsequent researchers which will be discussed below.

Many subsequent researchers have conducted investigations to arrive at better relationships for wind induced heat transfer coefficients. The experiments may be divided into four categories;

1) classical heat transfer experiments on flat plates of various geometries in wind tunnels.

2) Experiments on full sized solar collectors conducted outdoors.

3) Experiments on small mock-solar collectors or hot plates mounted outdoors.

4) Scaled experiments of solar collectors under the influence of a simulated atmospheric boundary layer in wind tunnels.

The experiments have investigated flow patterns, heat transfer coefficients, and their interactions. A survey of the work does not reveal conclusive results, but the relationships for wind induced heat transfer
coefficient, \( h \), are reviewed below by author.

Reviewing the translation by Serré (35) of the Nusselt and Jürges original paper, Nusselt and Jürges recommend:

\[
h_w = 7.14 V^{0.78} + 5.35 e^{-0.6V} \text{ W/m}^2\text{C} \tag{46}
\]

for \( 0 < V < 25 \text{ m/sec} \). They suggest that for practical applications the following equations may be used for:

\[
\begin{align*}
V < 5 \text{ m/sec}, & \quad h_w = 5.8 + 3.95V, \text{ W/m}^2\text{C} \tag{47} \\
V > 5 \text{ m/sec}, & \quad h_w = 7.14 V^{0.78}, \text{ W/m}^2\text{C} 
\end{align*}
\]

Watmuff et al., (36), suggest that if radiation effects are eliminated from Jürges relation (32) then a better relation would be:

\[
h_w = 2.8 + 3.0 V \text{ W/m}^2\text{C} \tag{48}
\]

for: \( 0 < V < 7 \text{ m/sec} \).

This justification is not valid as radiation effects were accounted for in the Jürges experiment.

The more recent work by Tien and Sparrow, 1977 (37), presents wind tunnel experiments for square plates, situated at various orientations to the flow direction. These experiments were performed by sublimation of naphthalene from the surface of a plate, and using an analogy between heat transfer and mass transfer they determined heat transfer coefficients. They concluded that, due to three-dimensional flow effects that heat transfer was almost independent of flow direction. These results were correlated to:

\[
j = 0.931 \text{ Re}^{1/2} \tag{49}
\]
where

\[ j = \left( \frac{h_w}{\rho C_p v_w} \right) Pr^{2/3} \]

where \( \rho \) is the density of the air,

\( C_p \) is specific heat,

\( Pr \) is the Prandtl number,

\( v_w \) is the free stream velocity of the air flow

\( Re \) is Reynolds number based on edge length.

This experiment was conducted on square plates and the authors suggest that it is not applicable to rectangular plates. Reynolds numbers from 20,000 to 100,000 were investigated.

Tien and Sparrow conclude that Jürges' relationship substantially overestimates the value of \( h_w \).

In a later work Tien and Sparrow, 1979 (38), discuss further experiments on the distribution of heat transfer coefficient across a plate. They once again state the insensitivity to wind direction and note that the highest heat-transfer coefficients occur near the edges of the plate. Flow patterns were observed to be highly complex and three dimensional. They note that flow patterns are not very dependent on Reynolds number. In Sparrow and Nelson's work, 1981 (39), the authors investigated wind related heat losses on leeward facing and windward facing collector surfaces using techniques similar to those used in their previous experiments. They concluded that leeward wind losses are slightly smaller, (i.e., 10%), than windward values, when \( Re \) is less than 60,000. For \( Re \) greater than 60,000 leeward values are higher. Flow visualization showed that eddies formed over the leeward surface and tended to enhance heat transfer. For
rectangular plates Sparrow and Tiens relation (40) is given as:

\[ \text{Nu} = 0.86 \, \text{Re}^{\frac{1}{3}} \, \text{Pr}^{1/3} \]  

(50)

where the characteristic length is four times the plate area divided by the plate perimeter.

The conclusions of the work of Sparrow is that eqn. (45) overpredicts \( h_w \) by two to three hundred percent and that wind direction has little effect.

Hewitt and Griggs (41) investigated wind effects on solar collectors in a mammoth study employing outdoor tests on real solar collectors, indoor experiments on mini-collectors and wind tunnel experiments on hot plates. The results are neither conclusive nor convincing.

The authors conclude that Jürges' results predict too high a heat loss at high velocities and too low a value at low velocities while Sparrow and Tien (37) uniformly overpredict losses by 10%.

Duffie and Beckman (40) suggest for collectors on buildings, that for \( h_w \), to use the larger of 5 W/m\(^2\)C or

\[ h_w = \frac{8.6 \, V^{0.5}}{L^{0.4}} \]  

(51)

where \( L \) is the cube root of the house volume. The value of 5 W/m\(^2\)C is suggested to be representative of the natural convection heat transfer case.

Green (42) suggests, that for velocities parallel to the collector surface
\[ h_w = 3.0 + 7.4 v_p^{1/4} \]  

These values were determined by tests on full scale collectors in a solar simulator, blowing wind from a "Scroll" fan, along the plate surface. The characteristics of the flow were not stated. They cite the work of Oliphant (43), to conclude that free stream wind velocity is 1.3 to 3 times the velocity parallel to the plate.

A review of the work of Oliphant suggests that his method for measuring velocity parallel to the collector glazing was invalid as the wind velocity was measured parallel to the ground and not parallel to the collector surface.

Hamelin (44) has illustrated the effect of \( h_w \) on solar collector performance, and calculates the natural convection heat transfer rate using the results of Fujii (45). Results show that the natural convection coefficient is a function of plate temperature, but values approach 4 W/m²K.

In a recent work of Kind et al (46), wind tunnel and full scale experiments were performed, attempting to maintain a turbulence intensity similar to natural wind. It is suggested that the turbulence intensity of wind has a significant effect on \( h_w \). The atmospheric boundary layer was modelled for the wind tunnel portion of the research.

It was concluded that heat transfer coefficient was independent of wind direction except for winds from the sides of the collector array. The wind tunnel results are a strong function of Reynolds number. The results also showed the occurrence of separation for leeward flows. When the wind
tunnel results of Kind et al (46) are extrapolated to full scale results a tentative conclusion is that Jürges' results are high.

None of the relations presented for wind induced heat transfer coefficient are conclusive and it was decided for this study, that for the theoretical evaluation of wind effects on solar collectors, values of \( h_w \) would be presented rather than wind velocity. This would ensure that the results would be applicable to future studies of the relationship between \( h_w \) and free stream wind velocity. The value of \( h_w = 5 \text{ W/m}^2\text{C} \) was chosen to represent the natural convection case ranging to values of \( 40 \text{ W/m}^2\text{C} \) for high \( V \).

One conclusion of the above described studies, is that wind direction does not seem particularly important as the heat transfer rate remains constant in the separation regions formed by leeward flow direction.

The relationship between \( h_w \) and free stream velocity is not well established particularly when the flow is influenced by obstructions. This was the case for the experimental portion of this study as the test collector was mounted on top of a building.

4.2.3.3 Convective Heat Transfer Coefficient, \( h_{c,p-c} \)

Natural or free convective heat transfer in a solar collector occurs between the absorber plate and the cover plate. The rate of heat transfer depends upon many factors, including plate spacing, tilt angle of the collector, air temperature and plate length.

Correlations for predicting convective heat transfer between inclined parallel plates have been presented, (48, 49, 50, 51). Buchberg et al (48)
reviewed the literature on natural convection in 1974. The work of Hollands et al (50, 51, 52, 53) is generally considered the most representative for predicting convective heat transfer coefficients in solar collectors. An excellent discussion of convective heat transfer between inclined air layers, heated from below, is presented in reference (53), and is summarized below.

When a fluid is stationary, (i.e., not undergoing convection), it transfers heat by conduction just as a solid and the heat transfer rate for a parallel, stationary, air layer, \( q_{st} \), between the absorber plate and cover of a solar collector is:

\[
q_{st} = \frac{A \cdot k_a \cdot (T - T_c)}{L_{p-c}}
\]

where:
- \( k_a \) is the thermal conductivity of the air,
- \( T, T_c \) are the temperatures of the absorber plate and cover plate,
- \( L_{p-c} \) is the thickness of the fluid layer, referring to figure (12).
The heat transfer coefficient for the stationary air layer, \( h_{st} \), is given:

\[
    h_{st} = \frac{A \frac{q}{st}}{T_p - T_c} = \frac{A \frac{k}{p \cdot a}}{L_p - c}
\]  

(54)

When a temperature difference is applied across the air layer a temperature variation is initially established by conduction, this then results in a density variation in the fluid. The resultant buoyancy force causes an imbalance between the gravity and hydrostatic pressure forces on the fluid resulting in a flow of the fluid. Under this convective flow thermal energy is transported by bulk movement of the fluid as well as conduction. Free convective heat transfer increases the heat flow relative to the stationary case. The Nusselt number, \( Nu \), represents the ratio of the heat transfer across the fluid layer in the convecting case relative to that in the purely
stationary, conducting case.

\[
Nu = \frac{h_c}{h_{st}} \left( \frac{L_p}{L_{p-c}} \right) = \frac{h_{c,t}}{h_{st}} = \frac{q_{conv}}{q_{st}}
\]  \hspace{1cm} (55)

The free convective motion and heat transfer are a function of a number of variables. These include the slope and temperature of the inclined surfaces, the length of the enclosure, \(L_p\), the distance between the plate and cover, \(L_{p-c}\), and the fluid properties which vary with temperature.

It can be shown, that for a fixed geometry and orientation, the total effect on free convective heat transfer as represented by \(Nu\) can be characterized by the Rayleigh number, \(Ra\), and the Prandtl number \(Pr\).

\[
Ra = \frac{C_p \rho^2 g \beta (T_p - T_c)}{\mu k_a} \frac{X_c}{\mu k_a}
\]

where \(g\) is the local acceleration of gravity, \(m/s^2\),

\(\beta\) is the volumetric thermal expansion coefficient of the fluid, \(K^{-1}\),

\(X_c\) is a characteristic dimension, \(L_{p-c}\) for the solar collector case, \(m\)

\[
Pr = \frac{C_p \mu}{k_a}
\]

Values for \(Ra\) for air are given (53);

\[
Ra = 2.37 \left( 1 + \frac{200}{T_{alm}} \right)^2 \left( \frac{100}{T_{alm}} \right)^4 \left( \frac{T_p - T_c}{100} \right) \left( \frac{L_{p-c}}{100} \right)^3 \left( \frac{p_{atm}}{\mu} \right)^2
\]  \hspace{1cm} (56)

where \(T_{alm}\) is the mean temperature of the fluid layer in \(K\), \(L_{p-c}\) is in meters and the atmospheric pressure, \(P_{atm}\), is in atmospheres.
For horizontal air layers where the thickness is much less than either lateral dimension, if the Raleigh number is less than a critical value of 1708, the fluid remains stationary due to viscous damping overcoming buoyancy effects. Thus heat transfer is by conduction only. At slightly higher Ra, instability causes a cellular motion to form in the fluid. The value of Nu increases rapidly with Ra from this point.

For horizontal surfaces with aspect ratio, (i.e., length of the plate divided by thickness of the air layer), greater than 10, Nu is only a function of Ra. For inclined surfaces, (slope = a, s > 0°), the value of Nu becomes increasingly dependent on aspect ratio with s, but for s < 70° and aspect ratio greater than 25 the dependence on aspect ratio is small. For s > 0 and if Ra is less than a critical value of 1708/cos s, "base" flow exists. Base flow is shown in figure (13) where a single cell is formed with fluid rising along the hot surface and falling along the cold one. In this case heat transfer is normal to the fluid flow and is subsequently by conduction only, i.e., Nu = 1. For Ra > 1708/cos s convective cells set in and Nu increases.

The relationship of Hollands (50, 52, 53) is most widely used to relate Nu as a function of s, for s = 0° to 75°, and aspect ratios greater than 25. Hollands recommends eqn. 57 for S=0° to 75°, but indicates that errors of 10% may occur for values of S between 60° and 75°.

$$\text{Nu} = 1 + 1.44 \left[ 1 - \frac{1708}{\text{Ra} \cos s} \right] + \left[ 1 - \left( \frac{\sin 1.8 \, s}{\text{Ra} \cos s} \right)^{1.6} \cdot 1708 \right] + \left[ \left( \frac{\text{Ra} \cos s}{5830} \right)^{1/3} - 1 \right]^{+}$$ (57)
where the meaning of the "+" exponent is that only positive values of the terms in the square brackets are to be used (i.e., use zero if the term is negative). Plots of \( \text{Nu vs Ra} \) for values of \( s \) from 0° to 75° are shown in figure (14) from reference (40). The effects of aspect ratio and geometry for varying values of \( s \) have been shown by Hollands (51). His results show that for solar collector applications \( \text{Nu} \) is dependent on the air layer thickness, \( L_{p-c} \), and that heat transfer may be increased or decreased by small variations in cover to absorber spacing, \( L_{p-c} \).

\[ \text{CORRELATION OF HOLLANDS ET AL. (1976)} \]

\[ \text{CONDUCTION LIMIT} \quad \text{Nu} \cdot 1 \]

**Figure 14**

NuSelt Number as a function of Rayleigh Number for free convection heat transfer between parallel flat plates at various slopes (Ref. 40)
Consider the case for the Chamberlain collector at 45° tilt, plate temperature \( T_p = 330 \text{ K} \), cover temperature \( T_c = 300 \text{ K} \), \( L_{p-c} = 0.019 \text{ m} \) and
\[
\frac{L_p}{L_{p-c}} = \frac{2.09}{0.019} = 110.0. \text{ Thus aspect ratio} = 110, \text{ i.e., greater than 25 so eqn. (57) holds. The value of \( Ra \) is;}
\[
Ra = 2737 \times (1 + \frac{200}{315})^2 \times (\frac{100}{315})^4 \times (330-300) \times (100-0.019)^3 1.0^3
\]
\[
= 15,290 \times 1.53 \times 10^4
\]
where \( T_{alm} = \frac{T_p + T_c}{2} = \frac{330 + 300}{2} = 315 \text{ K} \)
and \( P_{atm} = 1.0 \text{ atmosphere} \)

From eqn (57)
\[
\text{Nu} = 1 + 1.44 \times \left[ 1 - \frac{1708}{1.53 \times 10^4 \cos 45^\circ} \times (1 \frac{\sin(1.8 \times 45)}{1.53 \times 10^4 \cos 45^\circ}) \right] + \left[ \frac{1.53 \times 10^4 \times \cos 45^\circ}{5830} \right]^{1/3} - 1
\]
\[
= 1 + 1.44 \times [0.842]^\dagger + 0.845 + [0.229]^\dagger
\]

Note: all the terms are positive so all will be used in the calculation
\[
\text{Nu} = 2.25
\]
The value of the convective heat transfer coefficient, \( h_{c,p-c} \) is given by

\[
\text{eqn. (55)}
\]
\[
h_{c,p-c} = \frac{k_a \text{Nu}}{L_{p-c}}
\]

(24)
The value of the thermal conductivity for air is derived by the relationship (53);
\[
\begin{align*}
  k_a &= 0.002528 \cdot \frac{T_{\text{alm}}}{(T_{\text{alm}} + 200)} \quad \text{(58)}
\end{align*}
\]

For this example, \(T_{\text{alm}} = 315\text{K}\)

\[
\begin{align*}
  k_a &= 0.002528 \cdot \frac{315^{1.5}}{315 + 200} \\
    &= 2.74 \times 10^{-2} \text{ W/m K}
\end{align*}
\]

The value for \(h_{c,p-c}\) is:

\[
\begin{align*}
  h_{c,p-c} &= \frac{2.7 \times 10^{-2} \times (2.25)}{0.019} \\
            &= 3.25 \text{ W/m}^2\text{C}
\end{align*}
\]

It should be noted that this value is for convective heat transfer and does not include the radiative heat transfer between the absorber plate and cover. The total heat transfer coefficient for the plate to absorber is given by eqn. (26)

\[
U_{p-c} = h_{c,p-c} + h_{r,p-c} \quad \text{(26)}
\]

where, \(h_{r,p-c}\) is given by eqn. (22). For the case of the Chamberlain collector with plate emittance equal to \(\varepsilon_p = 0.12\) and cover emittance, \(\varepsilon_c\), equal to 0.88. The value of \(h_{r,p-c}\) for the previous example is,

\[
\begin{align*}
  h_{r,p-c} &= 5.67 \times 10^{-8} \frac{(330^2 + 300^2)(330+300)}{(1/0.12) + (1/0.88) - 1} \\
            &= 0.839 \text{ W/m}^2\text{C}
\end{align*}
\]

and \(U_{p-c} = 3.25 + 0.839 = 4.09 \text{ W/m}^2\text{C}\)

It is interesting to note that radiative exchange accounts for only 22% of the heat transfer in this case. Radiative exchange is a function of the
difference of the 4th power of absolute temperatures between the cover and absorber, and the emittance of the plate and cover. The Chamberlain collector has a "selective absorber surface". That is its optical properties are wavelength, $\lambda$, dependent. It is designed to have a high absorptance, $\alpha_p$, over the short wavelength regions characteristic of solar radiation and to have a low emittance, $\varepsilon_p$, for the longer wavelength associated with low temperature thermal radiation. The low $\varepsilon_p$ of the Chamberlain collector limits radiation heat transfer. If a non-selective absorber plate coating such as black paint was used ($\varepsilon_p = 0.95$) the value of $h_{r,p-c}$ would be:

$$h_{r,p-c} = 5.98 \text{ W/m}^2\text{C}$$

and

$$U_{p-c} = 5.98 + 3.25 = 9.23 \text{ W/m}^2\text{C}$$

For the non-selective absorber plate, (i.e., $\varepsilon_p = 0.95$) the heat transfer coefficient is increased 125% and now radiative heat transfer represents 65% of the heat transfer.

4.3 Transmission of Glazing and Effective Transmission-Absorption Product $(\tau\alpha)_e$.

The relationships for the transmission and absorption of glazing materials have been presented (1, 30, 52, 54).
4.3.1 Transmission of Glazings

For surfaces that are transparent to incident radiation to any degree, the sum of absorptance, $\alpha$, reflectance, $\rho$, and transmittance, $\tau$, must be unity; that is the incident radiation is absorbed, reflected and transmitted, figure 15. All three properties, $\alpha$, $\rho$ and $\tau$ are functions of the wavelength and angle of incidence of the incoming radiation as well as the refractive index, $n$, and the extinction coefficient, $K$, of the material. The values of $n$ and $K$ are also functions of wavelength, but for the narrow
spectrum of solar radiation they may be treated as being independent of wavelength (40). The expressions for the absorptivity, reflectivity and transmissivity for a glazing cover are presented below.

The refraction angle of incoming radiation is given by Snell's Law:

\[ \sin \theta = n \sin \theta_2 \]

or

\[ \theta_2 = \arcsin \left( \frac{\sin \theta}{n} \right) \] (59)

where \( n \) = index of refraction for the cover material

\( \theta \) = angle of incidence of incoming solar radiation

\( \theta_2 \) = refraction angle of solar radiation

For an air-glass interface the reflectivity is given by Fresnel's formulae. The reflectance has two components corresponding to the two components of polarization resolved parallel and perpendicular to the plane of incidence. Thus we have:

\[ \rho_{\parallel} = \frac{\tan^2(\theta - \theta_2)}{\tan^2(\theta + \theta_2)} \] (60)

\[ \rho_{\perp} = \frac{\sin^2(\theta - \theta_2)}{\sin^2(\theta + \theta_2)} \] (61)

The fraction of radiation that is absorbed in a single pass through the cover of thickness \( L_c \) is given by Bouger's law, i.e.:

\[ a_c = 1 - e^{-KL} \] (62)

where \( L \) is the optical path length given by;

\[ L = L_c / \cos \theta_2 \] (63)

so,

\[ a_c = 1 - \exp \left( \frac{-KL}{\cos \theta_2} \right) \] (64)
The transmission of the glazing due to absorption of the glazing only is then given by;

\[ \tau_a = e^{\left(-\frac{KL}{c}\right)\cos \theta_2 \tau_a} \]  
(65)

The total values of reflectance for each component of the radiation can be attained by the solution to Stoke's equations for the infinite series of reflections in the cover plate (55). These result in;

\[ \rho_{\perp, c} = \rho_{\perp} \cdot \left(1 + \frac{\tau_a^2 (1-\rho_{\perp})^2}{1 - \rho_{\perp}^2 \tau_a^2}\right) \]  
(66)

\[ \rho_{\parallel, c} = \rho_{\parallel} \cdot \left(1 + \frac{\tau_a^2 (1-\rho_{\parallel})^2}{1 - \rho_{\parallel}^2 \tau_a^2}\right) \]  
(67)

The values for transmittance are;

\[ \tau_{\perp, c} = \tau_a \cdot \left(\frac{(1-\rho_{\perp})^2}{1 - \rho_{\perp}^2 \tau_a^2}\right) \]  
(68)

\[ \tau_{\parallel, c} = \tau_a \cdot \left(\frac{(1-\rho_{\parallel})^2}{1 - \rho_{\parallel}^2 \tau_a^2}\right) \]  
(69)

The average values of transmittance and reflectance are then given by;

\[ \rho_c = \frac{\rho_{\perp, c} + \rho_{\parallel, c}}{2} \]  
(70)

\[ \tau_c = \frac{\tau_{\perp, c} + \tau_{\parallel, c}}{2} \]  
(71)

These relations apply to the single cover case, but relationship for multiple covers are presented in references (19, 30, 40) for covers of the
same material and thickness. Edwards (57) has presented relationships for multiple glazings of different materials and thickness.

\[ n = 1.518 \]
\[ K_L c = 0.035 \]

**Figure 16**
Transmission of Glass vs Beam Radiation Incident Angle

The values of transmittance shown in figure (16) are for beam radiation. Values of transmittance for the cover of the Chamberlain collector at various angles of incidence of beam radiation are shown in figure (16) as calculated by the above relationships utilizing the subroutine OPTCAL described in section 4.6.

The transmittance of diffuse solar radiation, \( \tau_d \), is usually evaluated by:

\[
\tau_d = \int \tau_c(\theta) \cdot \sin 2\theta d\theta \quad (72)
\]
assuming a uniform distribution of diffuse radiation across the sky. In the past it has been assumed that diffuse radiation reflected from the ground and different portions of the sky is the same, i.e., independent of tilt angle of the collector, resulting in a value of $\tau_d$ equal to the value of $\tau$ at 60° incidence angle when eqn. 72 is solved. Recently Brandemuehl and Beckman (56) have showed that diffuse radiation reflected from the ground should be treated differently from that coming from the sky. Figure (17) summarizes the work, where an effective incidence angle, $\theta_e$ is given for diffuse radiation from the ground and the sky as a function of collector slope, $S$.

![Diagram](image-url)

**Figure 17**

*Effective Incidence Angle of Isotropic Diffuse Solar Radiation from Brandemuehl and Beckman (1980)*
These results are applicable to glazing system of one and two covers with index of refraction, \( n \), between 1.34 and 1.526 and extinction lengths, \( KL_c \) less than 0.0524. Relationships for the results are (56),

For ground source diffuse, the effective incidence angle is

\[
\theta_{e, dg} = 90 - 0.5788(S) + 0.002693 \left( S \right)^2
\]  

(73)

for sky source diffuse, the effective incidence angle is

\[
\theta_{e, ds} = 59.68 - 0.1388(S) + 0.001497 \left( S \right)^2
\]  

(74)

If we define;

- \( \tau_{d, s} \) = the transmission of the glazing at an angle equal to the effective beam radiation incident angle, \( \theta_{e, ds} \),

- \( \tau_{d, g} \) = the transmission of the glazing at an angle equal to the effective beam radiative incidence angle, \( \theta_{e, dg} \),

- \( \rho_g \) = the reflectance of the ground adjacent to the solar collector for solar radiation,

- \( SSF \) = shape factor from cover surface at slope \( S \) to the sky;

\[
SSF = \frac{1 + \cos S}{2}
\]

- \( GSF \) = shape factor from cover surface at slope \( S \) to the ground:

\[
GSF = \frac{1 - \cos S}{2}
\]

the effective transmittance of the collector to diffuse radiation, \( \tau_{e, d} \) is then;

\[
\tau_{e, d} = \frac{\tau_{d, s} (SSF) + \rho_g (\tau_{d, g}) GSF}{SSF + \rho_g (GSF)}
\]  

(75)
4.3.2 Effective Transmittance-Absorptance Product

Of the radiation passing through the cover and striking the absorber plate of a solar collector, some is reflected back to the cover system. All this reflected radiation is not lost because some is reflected back to the absorber plate from the underside of the cover. In order to evaluate the resultant energy gain to the collector the transmittance-absorptance product, \((\tau a)\) must be evaluated. This is given as (1):

\[
(\tau a) = \frac{\tau \cdot \alpha_p}{1 - (1 - \alpha_p) \rho_d}
\]  
(76)

where \(\rho_d\) is the diffuse reflectance, and can be estimated by using the beam reflectance of the cover at an incidence angle of 60°. If part of the incoming solar radiation is diffuse then this value should be calculated for the diffuse radiation from the ground and sky, where \(\tau_{e,d}\) is the effective transmittance of the cover to diffuse sky and ground source solar radiation, as well as for beam solar radiation giving:

\[
(\tau a) = \frac{\tau_{e,d} \cdot \alpha_p}{1 - (1 - \alpha_p) \rho_d}
\]

(77)

and

\[
(\tau a)_b = \frac{\tau_c \cdot \alpha_p}{1 - (1 - \alpha_p) \rho_d}
\]

(78)

The derivation of \(U_L\) presented in section 4.2.3 assumed that the cover did not absorb solar radiation. In order to maintain the simplicity of that analysis and account for the reduced losses due to absorption of solar radiation by the glass an effective transmission-absorptance product \((\tau a)_e\) is used. All of the solar radiation absorbed by the cover is not lost.
as it tends to increase the cover temperature and hence, reduces losses from
the collector. This reduction in collector losses can be considered as an
increase in the value of the transmission-absorptance product. This is
given for the diffuse and beam components, (40);

\[
(\tau a)_{e,d} = (\tau a)_{d} + \alpha_{c,d} (1 - \frac{U_{T}}{U_{d} - \alpha_{b}})
\]  

(79)

\[
(\tau a)_{e,b} = (\tau a)_{b} + \alpha_{c,b} (1 - \frac{U_{T}}{U_{b} - \alpha_{c}})
\]  

(80)

where \( \alpha_{c} = (1 - \tau_{a}) \) for the diffuse and beam components is
determined respectively.

The effective transmittance-absorptance product for both beam and diffuse is
given by;

\[
(\tau a)_{e} = [(1-f_{d})(\tau a)_{e,b} + f_{d} (\tau a)_{e,d}]
\]  

(81)

where: \( f_{d} \) is the fraction of the total incident solar radiation on the
collector that is diffuse from both the sky and ground.

4.4 Heat Removal from Flat-Plate Solar Collectors

The analysis of heat loss coefficient described in section 4.2
relates the heat loss from the solar collector to the mean absorber plate
temperature, \( T_{pm} \).

As fluid entering the collector at a temperature, \( T_{fi} \), moves along
the absorber plate, heat is transferred to it, (assuming \( \eta \approx 0 \)). This results
in an increase in the fluid temperature and a subsequent increase in the
absorber plate temperature. This increased absorber plate temperature
results in increased heat loss $q_L$. The distribution of $T_r$, and subsequent $T_{pm}$ through the collector is usually non-linear. The temperature profile along the length of the absorber plate is a function of the fin efficiency of the collector plate, $F$, the fluid mass flow rate, $m$, the fluid to tube wall heat transfer coefficient, $h_w$, and the local fluid temperature. The resultant value of $T_{pm}$ is therefore difficult to determine. As described in section 2, Bliss (3) developed factors, $F'$, collector efficiency factor, and $F_R$, collector heat removal factor, so that collector heat loss could be related to inlet fluid temperature, $T_{fi}$. In designing solar heating systems, $T_{fi}$ is relatively easy to predict. To maintain the integrity and simplicity of the Hottel, Whiller, Bliss collector model (described in section 2), the concepts of $F_R$ and $F'$ will be utilized for this analysis of collector performance.

4.4.1 Collector Heat Removal Factor $F_R$

The heat removal factor, $F_R$ is the ratio of actual useful energy gain of the collector to the useful gain if the whole collector absorber plate surface was at the fluid inlet temperature, $T_{fi}$.

$$F_R = \frac{m C_p (T_{fo} - T_{fi})}{A_c \left[ \frac{(\tau_0) T_i - U_L (T_{fi} - T_a)}{} \right]}$$

The concept of $F_R$ is analogous to conventional heat exchanger effectiveness, and has been shown (1, 3, 22, 40) to be given by;

$$F_R = \frac{G C_p}{U_L} \left[ 1 - e^{-F'U_L/(GC_p)} \right]$$

(83)

where $G$ is the specific mass flow rate.
\[ G = \frac{m}{A_c} \]  

(84)

The useful energy collected in the solar collector is then;

\[ q_u = A_c F_R \left( \frac{e}{\eta} I - U_L (T_{fi} - T_a) \right) \]  

(85)

The factor \( F_R \) reduces the calculated useful energy gain from what it would have been had the whole collector been at \( T_{fi} \), to what it actually is for a fluid that increases in temperature along its flow path. As mass flow rate through the collector increases, the temperature rise through the collector decreases, causing lower heat losses since \( T_{pm} \) is reduced. As the flow rate becomes very large, the temperature rise from inlet to outlet approaches zero, but the absorber plate surface temperature must still be higher than the fluid temperature for heat transfer. This temperature is accounted for by the collector efficiency factor \( F' \). It may be seen that \( F_R \) is dependent on the mass flow rate, heat loss coefficient \( U_L \), \( F' \), and \( C_p \).

4.4.2 Collector Efficiency Factor, \( F' \)

The collector efficiency factor, \( F' \), is a measure of the ability of the collector to transfer heat to the fluid. Equations for \( F' \) for various collector designs have been derived by numerous authors (2, 3, 22, 40). \( F' \) represents the ratio of the actual useful energy gain to the useful energy gain that would result if the collector absorbing surface were at the mean fluid temperature \( T_{pm} \). It may also be thought of as the ratio of the heat transfer resistance from the absorber plate to the ambient air, \( 1/U_L \), to the heat transfer resistance from the fluid to the ambient air, \( 1/U_o \). Thus;
\[
F' = \frac{1}{U_L} \frac{1}{U_o}
\]

or

\[
F' = \frac{U_o}{U_L}
\]

the value of \( U_o \) can be shown (40) to be given by:

\[
U_o = \left[ W \left( \frac{1}{U_L} \left( \frac{1}{D+(W-D)F} \right) + \frac{1}{C_b} + \frac{1}{mD_i h_f} \right) \right]^{-1}
\]

(88)

where \( W, D \) - are physical dimensions of the absorber plate referring to figure (18).

\( F \) - is the fin efficiency of the absorber plate.

\( C_b \) - is the bond conductance from the absorber plate to tube.

\( D_i \) - is the inside tube diameter.

\( h_f \) - is the fluid to tube heat transfer coefficient.

For most modern solar collectors, the tube to plate bond conductance is high so the term \( 1/C_b \) approaches zero.

FIGURE 18
Tube in sheet absorber plate,
Dimensions for Eqn. 88
4.4.3 Fin Efficiency, $F$

Most conventional flat plate solar collectors are constructed with a "tube in/on sheet" absorber plate similar to that shown in figure (18). Values of fin efficiency, $F$, are derived for many absorber plate designs (1, 3, 22) but generally values as high as practically possible are chosen. For the case shown in figure (18), the fin efficiency is given by:

$$F = \frac{\tanh \left( \frac{X(W-D)/2}{X(W-D)/2} \right)}{X(W-D)/2}$$

(89)

where

$$X = \left[ \frac{U_L}{(k_p)_{f}} \right]^{\frac{1}{4}}$$

(90)

The value of $F$ is dependent on the heat loss coefficient of the solar collector, $U_L$, the thermal conductivity of the absorber plate $k_p$, and $\delta$, the fin thickness.

4.4.4 Heat Transfer Coefficient, $h_f$

The rate of heat transfer from the tube wall to the fluid is characterized by the heat transfer coefficient, $h_f$, which is dependent on Reynolds number and Prandtl number for the flow situation. Reynolds number is given by:

$$Re = \frac{V D_h}{v}$$

(91)

where $V$ is the velocity of the fluid flowing in the tubes, $v$ is the Kinematic viscosity of the fluid and $D_h$ is the hydraulic diameter of the tubes, i.e.,
Prandtl number is given by:

\[ Pr = \frac{\mu C_p}{k_f} \]  

(93)

where \( \mu \) is the dynamic or absolute viscosity of the fluid, and \( k_f \) is the fluid thermal conductivity.

The above properties should be evaluated at the mean fluid temperature.

The flow in solar collectors is usually low and subsequently is laminar. The tube length is usually short so flow is rarely fully developed.

Many relationships for determining \( h_f \) exist (19, 22, 34) but may be separated into two categories; constant heat flux and constant wall temperature cases. Duffie and Beckman (40) have suggested that the heat transfer in solar collectors lies between the two cases. High loss collectors approach the constant wall temperature case and low heat loss collectors approach the constant heat flux condition. Generally, the heat transfer coefficients are very similar under either condition, with constant wall temperature values being slightly lower.

For laminar flow in the entrance region of pipes, (constant wall temperature case), the value of \( Nu_f \) (77) is given:

\[ \frac{0.0668 \ (RePrD_h/L)}{h} \left( 1 + 0.04 \ (RePrD_h/L)^{2/3} \right) \]

(94)
where;
\[ h_f = N_u f \left( \frac{k_f}{D_h} \right) \]  \hspace{1cm} (95)

4.4.5 Calculation of \( F \) for Chamberlain Collector

The absorber plate of the Chamberlain solar collector is not of conventional tube and sheet construction but consists of two steel sheets, "stitch" welded together then expanded to form flow passages. The header is integral to the absorber plate. This design results in a highly efficient absorber plate as much of the absorber surface is in direct contact with the heat transfer fluid, (except for the regions of the welds). For the purpose of analysis the absorber plate is represented by 19 parallel flow passages with the weld region forming a small fin between the flow passage. The dimensions of the Chamberlain absorber plate is shown in figure (19).

![Diagram of Chamberlain flow passage configuration](image)

**FIGURE 19**
CHAMERLAIN FLOW PASSAGE CONFIGURATION

The value of \( U_o \) can be estimated from,
\[ U_o = \left[ W \left( \frac{1}{U_{L}[W \cdot F]} + \frac{1}{W_{per} \cdot h_f} \right) \right]^{-1} \]  \hspace{1cm} (96)

The value of \( F \) is determined as 0.99, from manufacturers' data. The value of \( C_b \), bond conductance does not apply as the tubes are integral in the plate. \( W_{per} \) is the wetted perimeter of the tubes.
4.5 Calculation of Theoretical Thermal Efficiency

The rate of energy extraction by the solar collector heat transfer fluid is given by eqn (4);

\[ Q_u = A_c \cdot F_R \left[ I(\tau_0) e - U_L (T_{fi} - T_a) \right] \]  \hspace{1cm} (4)

The collection efficiency is the ratio of \( Q_u \) to the energy incident on the collector reference surface area, \( A_c \), thus:

\[ \eta = \frac{Q_u}{A_c \cdot I} \]  \hspace{1cm} (97)

To calculate the theoretical efficiency of a solar collector by the relationships presented in the previous sections, an iterative procedure must be followed. While the value of absorbed solar radiation, \( I(\tau_0) \), can be calculated directly; the value of \( (\tau_0)_e \) is a function of \( U_L \) and \( U_L \) which is in turn a function of the mean absorber plate temperature, \( T_{pm} \). The mean absorber plate temperature is a function of the mean fluid temperature \( T_{fm} \), which is a function of \( F' \). The heat removal factor \( F_R \) is a function of \( U_L \).

To calculate theoretical thermal efficiency, first an estimate of the mean plate temperature is made. This value should be close to the inlet fluid temperature. From this estimated values of \( T_{pm} \), \( U_L \) and \( (\tau_0)_e \) are determined. The value of \( T_{fm} \) is estimated, \( (i.e., \ T_{fm} = T_{fi} \) and the value of \( F' \) and \( F_R \) are determined using the previously calculated value of \( U_L \). With these approximate values for \( F_R \) and \( U_L \) and the value of \( I(\tau_0)_e \), the value of \( Q_u \) is determined by eqn. (4).
The actual value of \( T_{fm} \) is determined by integrating the fluid temperature along the flow direction of the absorber plate, i.e.,

\[
T_{fm} = \frac{1}{L} \int_{0}^{L} T_f(x) \, dx
\]

This integral has been shown to be represented by (1);

\[
T_{fm} = T_{fi} + \frac{Q_{U/A}}{U_L FR} \left[ 1 - \frac{F_R}{F_f} \right]
\]

(98)

Using this relationship, new values of \( T_{fm} \) may be determined and \( F_R \) calculated in an iterative manner until convergence is obtained on \( T_{fm} \).

The value of \( T_{pm} \) may be estimated by (1);

\[
T_{pm} = Q_{U} \cdot R_{p-f} + T_{fm}
\]

where \( R_{p-f} \) is the heat transfer resistance between the plate and the fluid.

\[
R_{p-f} = \frac{1}{h_f \cdot W \cdot L \cdot \eta_t}
\]

thus

\[
T_{pm} = T_{fm} + \frac{Q_{U}}{h_f \cdot W \cdot L \cdot \eta_t}
\]

(99)

Using this new value for \( T_{pm} \), the value of \( U_L \) and \( T_{e} \) may be recalculated and process repeated until the value of \( T_{pm} \) converges.

During the calculations of \( U_L \) the value of \( U_{top} \) must be determined by an iterative calculation for each value of \( T_{pm} \). To do this the cover temperature is estimated (i.e., \( T_c = (T_m + T_a)/2 \), and \( U_{top} \) determined. A new value of \( T_c \) is then determined and the value of \( U_{top} \) recalculated until \( T_c \) converges.
The calculation of collection efficiency is then made by eqn (97) utilizing the calculated values of $(\tau a)_e$, $P_R$, and $U_L$ for the specific meteorological conditions.

These calculations are lengthy by hand so the calculations are performed by the computer subprogram COLSM, described in Section 5.
5. COMPUTER SIMULATION OF THERMAL PERFORMANCE

As described in section 4.5, the calculation of solar collector theoretical efficiency is an iterative process involving $T_{pm}$, $T_f$ and $T_c$. For this reason the calculations are most efficiently made on a computer.

The calculations presented in section 4.5, are performed by the subprogram COLSM. This program and the associated main and subprograms were written in Fortran IV, (59). The routines were written and data analyzed on a Digital Equipment Corp., D.E.C. LSI-11-03 digital computer. The computer system operated in single job monitor under RT-11SJ version 3B, (60). The system was configured with a RX01 dual floppy disk drive, (of 1 mega-byte capacity each), two RL01 rigid disk drives, (of 5 mega-byte capacity each), a Decwriter III hard copy terminal and a VT52 CRT. Data could also be transmitted to a Hewlett Packard 2647A intelligent graphics terminal and serial HP 7225A hard copy plotter, for plotting on the HP terminal screen or hard copy plotter. The HP terminal utilized graphics software issued from the D.E.C. Computer, (61).

While the computer system described above represented a very powerful tool for the analysis of data, the COLSM program will run, (minus the graphics capability), on a conventional LSI-11-03 D.E.C. computer, utilizing only the RX01 floppy disk drives and hard copy terminal.

The simulation program was designed and structured utilizing a number of subroutines to perform each step of the calculations. When these subroutines are compiled, they are "linked" together into a single routine of object code for execution by the computer.

The subroutine structure of the simulation allows for checking of
each portion of the simulation independently with changes made only to the particular subroutine being updated. This is a distinct advantage in large simulation programs as is the case for this program which solves in excess of 50 equations iteratively, for each operational point. An advantage of splitting the calculations into logical sections is that these subroutines can be run independently to evaluate the effect of operational and meteorological factors on each of the major components of the heat transfer analysis.

The analysis of component heat transfer and optical performance was done and is described in section 6.0. This analysis provides an insight into the heat transfer factors that affect the overall performance of the solar collector.

The input data, including collector characteristics, operational and meteorological factors are transferred to the simulation subprogram COLSM by the main driver routine RCOLSM. The main program RCOLSM also sets up the data output on the hard copy terminal and produces plots of thermal performance on the H.P. graphics terminal or H.P. hard copy plotter. The program runs interactively, allowing the user to rapidly see the effects of changing operational and meteorological factors.

The subprogram COLSM simulates the performance of the Chamberlain solar collector. It sets up the calculation of optical and heat transfer factors by the appropriate subroutines and allows the iterative calculations between subroutines. The subroutine COLSM makes calls to the subroutines TOPL, STAEFF and APFR which in turn make calls to the subroutines TSKY, CHTPC, OPTCAL and FLPROP as required. In all there are three levels of
subroutine calls from the main program RCOLSM.

The subroutine TOPL calculates the values of top heat loss coefficient, $U_{top}$. TOPL makes calls to the subroutine TSKY to calculate the effective sky temperature for the calculation of $h_r,s$ and $h_r,c,s$. TOPL also calls the subroutine CHTPC to calculate the convective heat transfer coefficient between the absorber plate and cover glass, $h_{c,p-c}$.

The subroutine STAEFF when called by the COLSM routine calculates the values of the effective transmittance x absorptance product, $(\tau\alpha)_e$, for specific conditions of collector slope, incident angle and $U_L$. The subroutine makes calls to the subroutine OPTCAL as required to calculate the optical performance of the cover.

The subroutine APFR when called by the COLSM routine calculates the value of the heat removal factor, $F_R$, for specific operational and meteorological factors. The APFR subroutine calls the subroutine FLPROP to calculate the heat transfer properties of the heat transfer fluid as required in the calculation of $F_R$ by the FLPROP routine. Each of the routines was checked thoroughly and calculated values for standard conditions compared to available literature to confirm their accuracy.

A short description and flow chart for each of the computer routines is provided in the following sections. Listings of the routines are provided in Appendix B. Results produced by RCOLSM and the subroutine TOPL, STAEFF and APFR are presented in Section 6.

5.1 COLSM Subroutine

The subroutine COLSM simulates the performance of the Chamberlain
flat-plate solar collector. The program will simulate other single glazed flat plate collectors if slight modifications are made. The value of \( U_o \) as calculated by the subprogram APFR is specific to the Chamberlain solar collector. This can be easily changed for more standard absorber plate configurations. Similarly, simulation of multiple cover collectors can be accomplished by modification to the subroutine TOPL and STAEEF to account for multiple glazings.

A flow chart of the COLSM routine is shown in figure (20) and listing provided in Appendix B. The subprogram COLSM specifies the collector optical and heat transfer characteristics, and the dimensions of the collector. It calls the appropriate subroutine to perform the heat transfer analysis and controls iterations between \( T_{pm} \) and \( T_{fm} \). COLSM solves eqn's; 16, 14, 98, 99, 97 and calls subroutine TOPL, STAEEF and APFR.

5.2 Subroutine TOPL

The subroutine TOPL calculates the values of top loss heat transfer coefficient, \( U_{top} \), for a single cover flat plate solar collector inclined at an angle to the horizontal for specific values of \( h_w \), \( T_{pm} \), \( T_{sky} \) and \( T_a \).

A flow chart of the routine is shown in figure (21), and a listing in Appendix B. The routine calls subroutines, \( T_{sky} \) to calculate an effective sky temperature for the calculation of \( h_{r,p-c} \) and \( h_{r,p-s} \). The TOPL routine calls the subroutine CHTPC to calculate \( h_{c,p-c} \) for specific collector slope, \( T_{pm} \) and \( T_c \).

The routine TOPL solves eqn's; 31, 33, 23, 26, 34, 35.
Figure 20

Flow chart for program to simulate the thermal performance of a flat-plate solar collector at specific operational and atmospheric conditions.
5.3 Subroutine TSKY

The subroutine TSKY calculates the effective sky temperature for a surface inclined at an angle "tilt" for specific values of $T_a$ and ground emissivity. It assumes the ground is at a temperature $T_g = T_a$. The shape factors between the surface and the sky and ground are calculated for a specific inclination angle and accounted for in the calculation. The routine solves eqn's; 41, 44, 43, 42. The flow chart for TSKY is presented in figure (22).

5.4 Subroutine CHTPC

The subroutine CHTPC calculates the convective heat transfer coefficient between the collector absorber plate and cover, $h_{c,p-c}$. The calculation is for specific inclination angle, $T_{pm}$ and $T_c$. A flow chart of CHTPC is shown in figure (23). The routine solves eqn's; 56, 58, 57, 24.

5.5 Subroutine STAEFF

The subroutine STAEFF calculates the effective transmittance $x$ absorbance product for specific cover/absorber combinations, at inclination angle, incident angle, fraction diffuse radiation, ground reflectance, and $U_L$. The routine solves eqn's; 73, 74, 75, 77, 78, 79, 80, 81, and makes 4 calls to the subroutine OPTCAL. Figure (24), shows the flow chart for STAEFF.
5.6 Subroutine OPTCAL

The subroutine OPTCAL calculates the optical performance, transmittance, reflectance and absorptance of a single cover using eqn's; 59, 60, 61, 65, 66, 67, 70, 68, 69, 71. The values are calculated for specific beam incident angle, \( n \) and \( K_{Lc} \). Figure (25), shows the flow chart for the OPTCAL routine.

5.7 Subroutine APFR

The routine APFR calculates the heat removal factor, \( F_R \) for a specific heat transfer, fluid, mass flow rate, and \( U_L \). The routine solves eqn's; 91, 94, 95, 96, 87, 83 and calls the subroutine FLPROP to calculate the heat transfer properties of the heat transfer fluid. Figure (26) shows the flow chart for APFR.

5.8 Subroutine FLPROP

The subroutine FLPROP (44) calculates the heat transfer properties for water or a 50/50\% mixture by volume of water and ethylene glycol for a specific \( T_{fm} \). A flow chart for FLPROP is given in figure (27).

5.9 Program RCOLOM

The main program RCOLOM specifies the meteorological conditions for a specific run of COLSM. It runs interactively with the user and calls subroutine PLTSTP to plot the results of a simulation run on either hard copy or graphics screen. This routine is specific to the H.P. equipment, but may only need minor modifications for other computer systems. Figure (28) shows the flow chart for RCOLOM.
SUBROUTINE TOPL

START

READ IN COLLECTOR CHARACTERISTICS, OPERATIONAL & ATMOSPHERIC CONDITIONS

LET $T_c = \frac{T_{pm} + T_a}{2}$

CALC. EFFECTIVE SKY TEMPERATURE

CALL TSKY

SUBROUTINE TSKY

CALC. $h_{r.c-a}$

CALC. $U_{c-a} = h_w + h_{r.c-a}$

CALL CHTPC

SUBROUTINE CHTPC

CALC. $h_{c.p.c}$

CALC. $T_c = T_{c,new}$

CALC. $U_{p.c} = h_{c.p.c} + h_{r.p.c}$

CALC. $U_{r.p.s}$

CALC. TOP HEAT LOSS COEFF. $U_{top} = \left[ \frac{1}{U_{p.c}} + \frac{1}{U_{c-a}} \right]^{-1} + U_{r.p.s}$

CALC. NEW COVER TEMPERATURE $T_{c,new} = \frac{T_{pm} + T_a}{U_{p.c}}$

IF $T_c > T_{c,new}$ THEN

OUTPUT VALUES

RETURN

FIGURE 21
FLOW CHART FOR SUBROUTINE TO CALCULATE TOP HEAT LOSS COEFFICIENTS
SUBROUTINE TSKY

START

READ IN
ATMOSPHERIC CONDITIONS
SURFACE INCLINATION

CALC. INCIDENT LONGWAVE
ATMOSPHERIC RADIATION
TO HORIZ. SURFACE R_s

CALC. ATMOSPHERIC LONGWAVE
RADIATION ON SLOPED SURFACE
R_s(S)

T_g - T_a

CALC. GROUND SOURCE
LONGWAVE INCIDENT ON
SLOPED SURFACE, R(si)

CALC. TOTAL LONGWAVE
ON SLOPED SURFACE, R(si)

CALC. EFFECTIVE SKY TEMP.
ON SLOPED SURFACE, R(si)

OUTPUT VALUES

RETURN

FIGURE 22
FLOW CHART FOR SUBROUTINE TO CALCULATE THE
EFFECTIVE SKY TEMPERATURE
SUBROUTINE CHIPC

START

FROM DRIVER
i.e. TOPM
READ IN COLLECTOR
CHARACTERISTICS, OPERATIONAL
& ATMOSPHERIC CONDITIONS

CALC. MEAN TEMP. OF AIR LAYER
\[ T_{\text{aim}} = T_{\text{pm}} + \frac{T_c}{2} \]

CALC. ABSORBER PLATE TO COVER
TEMP. DIFFERENCE, \( DT_{\text{al}} \)

CALC. RAYLEIGH NO. AT
\( T_{\text{atm}} \) & \( DT_{\text{al}} \)

CALC. NUSSELT NO.

CALC. CONVECTIVE HEAT TRANS.
COEFF. PLATE TO COVER
\[ h_{c,p-c} = \text{Nu} \left( \frac{K_a}{\rho_c} \right) \]

OUTPUT VALUES

RETURN

FIGURE 23
FLOW CHART FOR SUBROUTINE TO CALCULATE
CONVECTIVE HEAT TRANSFER COEFFICIENT FROM
THE ABSORBER PLATE TO COVER
FIGURE 24
FLOW CHART FOR SUBROUTINE TO CALCULATE THE EFFECTIVE, TRANSMITTANCE, ABSORPTANCE PRODUCT, $(r \alpha)$. 

SUBROUTINE STAEFF

START

FROM DRIVER ROUTINE I.e. COLSM

READ IN COLLECTOR CHARACTERISTICS, OPERATIONAL & ATMOSPHERIC CONDITIONS

CALC. TRANSMISSION OF GLAZING TO BEAM RADIATION $T_{C,B}$ -> CALL OPTCAL -> SUBROUTINE OPTCAL

CALC. EFFECTIVE INCIDENT ANGLE-SKY DIFFUSE

CALC. TRANSMISSION OF GLAZING TO SKY DIFFUSE -> CALL OPTCAL -> SUBROUTINE OPTCAL

CALC. EFFECTIVE INCIDENT ANGLE-GROUND SOURCE DIFFUSE

CALC. TRANSMISSION OF GLAZING TO GROUND SOURCE DIFFUSE -> CALL OPTCAL -> SUBROUTINE OPTCAL

CALC. TRANSMISSION OF GLAZING FOR GROUND AND SKY DIFFUSE

CALC. ABSORPTANCE OF GLAZING TO DIFFUSE

CALC. $(r \alpha)$ -> CALL OPTCAL -> SUBROUTINE OPTCAL

CALC. EFFECTIVE TRANSMITTANCE $\times$ ABSORPTANCE PRODUCT $(r \alpha)$

OUTPUT VALUES

RETURN
FIGURE 25
FLOW CHART FOR SUBROUTINE TO CALCULATE THE TRANSMITTANCE AND ABSORPTION FOR A SINGLE COVER GLASS
FIGURE 26
SUBROUTINE TO CALCULATE THE HEAT REMOVAL FACTOR FOR A FLAT-PLATE SOLAR COLLECTOR
SUBROUTINE FLPROP

START

FROM DRIVER
I.e. APFR

READ IN
FLUID TYPE
MEAN FLUID TEMP.

FLUID IS
WATER

FLUID IS 50%50%
WATER AND GLYCOL

CALC. \( C_p \)
CALC. \( \mu \)
DEN SITY
KINEMATIC VIS COSITY
AB Solute VIS COSITY

CALC. DENSITY
CALC. KINEMATIC
VIS COSITY
CALC. ABSOLUTE
VIS COSITY
CALC. PRANDTL NO.
\( Pr \)
OUTPUT VALUES
RETURN

VALUES FOR
WATER

FIGURE 27
FLOW CHART FOR PROGRAM TO CALCULATE HEAT TRANSFER FLUID PROPERTIES
PROGRAM RCOLSM

START

SPECIFY COLLECTOR CHARACTERISTICS

READ IN ATMOSPHERIC AND OPERATIONAL CONDITIONS

CALL COLSM

OUTPUT PERFORMANCE SUMMARY

CALL PLTSTP

OUTPUT PLOTTED RESULTS

OUTPUT RESULTS TO HARD BED PLOTTER

OUTPUT RESULTS TO GRAPHICS TERMINAL

CHANGE CONDITIONS

CHANGE CONDITIONS

User Input through CRT Terminal

Yes

RUN AGAIN

No

STOP

FIGURE 28
FLOW CHART FOR MAIN PROGRAM TO INPUT ATMOSPHERIC CONDITIONS AND PLOT SOLAR COLLECTOR PERFORMANCE CURVES
6. THEORETICAL RESULTS - EFFECTS OF METEOROLOGICAL AND OPERATIONAL FACTORS

The basic Hottel, Whillier, Bliss model for solar collector thermal performance has been presented in section 2.2. A basic assumption of this representation and associated test procedures (4), is that thermal and optical properties are constant. It is assumed that losses are dependent only on the temperature difference between the absorber plate and ambient air, surrounding the collector.

The problems experienced in ensuring these assumptions are met while performing thermal performance tests has been discussed in section 3. It has become apparent that obtaining repeatable, accurate test results is difficult.

Section 4 presents a detailed heat transfer analysis of the optical and heat transfer characteristics of a flat plate solar collector. Emphasis was placed on the meteorological factors that influence heat transfer and optical parameters.

The results here utilize the analysis described in section 4 and the computer programs described in Section 5. The effects of meteorological factors on the heat transfer of the Chamberlain, single glazed, flat plate, solar collector, (described in Appendix A) are presented. Results are presented for: the effects on $U_L$ through top loss heat transfer coefficient, $U_{top}$, the optical system through $(\tau_0)_e$, and the absorber plate heat transfer through $F_R$. These three components make up eqn (6) for the thermal performance of a flat plate solar collector.

$$
\eta = \frac{Q_{U}}{IAC} = F_R(\tau_0)_e - F_RU_L \left( \frac{f - T}{I} \right)
$$

(6)
6.1 Meteorological Effects on Heat Transfer in Flat-Plate Solar Collectors

Solution of the heat transfer analysis presented in section 4 allows the determination of the effects of meteorological factors on the heat transfer and optical processes in a flat plate collector.

6.1.1 Top Loss Heat Transfer Coefficient

The heat transfer through the cover of a flat plate solar collector is the major source of thermal losses. The analysis presented in section 4.2.3 demonstrates that the loss mechanism is a combination of conduction, convection and radiative heat loss from the absorber plate of the solar collector to the surrounding atmosphere. Eqn. (15) presents this heat loss as:

\[ Q_L = A_c U_L (T_{pm} - T_a) \]

where:

\[ U_L = U_e + U_T + U_b \]

The heat loss through the back and sides of most conventional solar collectors (and the Chamberlain collector) are small and relatively insensitive to changing meteorological conditions other than \( (T_{pm} - T_a) \). \( U_T \) (and \( U_{top} \)), are dependent on a variety of operational and meteorological factors.

A common assumption is that \( U_{top} \) is constant. Figure (29) shows values of \( U_{top} \) vs \( (T_{pm} - T_a) \). The value of \( (T_{pm} - T_a) \) is increased by effectively increasing the plate temperature, similar to the procedure described in the standard experimental test procedure (4).

It is obvious that \( U_{top} \) is in fact not constant but increases with
FIGURE 29
VARIATION IN TOP LOSS COEFFICIENT, $U_{\text{top}}$, FOR FIXED METEOROLOGICAL CONDITIONS

FIGURE 30
EFFECT OF COLLECTOR SLOPE
ON $U_{\text{top}}$ FOR $h_w = 5$ W/m$^2$ °C AND $T_s = T_a$

FIGURE 31
EFFECT OF $h_w$ ON $U_{\text{top}}$ FOR EFFECTIVE SKY TEMPERATURE = $T_a$

FIGURE 32
EFFECT OF COLLECTOR SLOPE AND $h_w$ ON $U_{\text{top}}$
increasing \((T_p^m - T_a^m)\). From the analysis presented in section 4 it may be
determined that as \(T_p^m\) is increased, the value of \(h_c, p-c\) increases. The
values of \(h_r, p-c\) and \(h_r, p-s\) increase as the difference of the fourth power
of absolute temperature rather than just linearly with \((T_p^m - T_a^m)\). As \(T_p^m\)
increases so does the value of \(T_{alm}\), the mean air layer temperature. This
results in a variation in the air properties and influences the convective
heat transfer through the \(Ra\) number. The value of \(Ra\) increases as \((T_p^m - T_a^m)\)
increases and consequently \(Nu\) and \(h_{c, p-c}\) increase.

The Chamberlain solar collector has a "selective" absorber plate with
emissivity to thermal radiation, \(\varepsilon_p\), equal to 0.12. Many collectors are
constructed with "non-selective" absorbers with much higher, \(\varepsilon_p\),
(i.e., \(\varepsilon_p = 0.95\)). In this case the variation in \(U_{top}\) becomes even more
significant.

As the difference between \(T_p^m\) and \(T_a^m\) becomes very small, heat
transfer by convection is reduced until at a value below the critical \(Ra\)
number, where heat loss is by conduction through the air layer,
(i.e., \(Nu = 1\)). The value of \(U_L\) remains relatively constant for smaller
values of \((T_p^m - T_a^m)\) after this point. This condition rarely occurs for
inclined solar collectors. The critical \(Ra\) number is given by \(Ra/cos S\),
where "S" is the inclination angle of the collector. Thus at high
inclination angles a larger value of \((T_p^m - T_a^m)\) is required to initiate
convection heat transfer.

The effect of collector slope on heat transfer is indicated in figure
(30) where \(U_{top}\) is presented for inclinations of 0, 45 and 75°, to the
horizontal. It may be seen that the value of convective heat transfer
decreases with increasing collector slope. The region indicated as "A" on the 75° tilt curve represents the region below the critical Ra number where heat loss is largely by conduction and convection has not started. These results correspond with the values of Nu vs tilt angle presented in figure (14) of section 4.

While the effect of tilt angle is largely an operational factor, during outdoor thermal performance testing, in order to meet the requirement for normal incidence to beam radiation from season to season, the solar collector must typically be adjusted in tilt through a range from 20 to 75°, (for the Ottawa latitude ~45°NLat.) If the collector being tested is mounted on a altitude/azimuth tracking test frame then the tilt may be steeper to obtain normal incidence in the morning or afternoon periods.

While tilt angle has a significant effect on $U_{top}$, the value of $U_{pc}$ represents only one component of the heat loss from the absorber. Another major resistance to heat transfer is provided by the film coefficient on the outer surface of the glass cover. This is referred to as, wind heat transfer coefficient, $h_w$, although the heat transfer may be that of natural convection if wind velocity is very low. The discussion in section 4 places a lower limit of $h_w = 4$ or 5 W/m²°C for this situation. Increasing wind velocity increases the value of $h_w$. The effect of increasing $h_w$ on $U_{w top}$ is presented in figure (31) for values of $h_w$ ranging from 5 to 30 W/m²°C.

As $h_w$ increases, it offers less resistance to heat loss from the absorber plate and consequently, $U_{top}$ increases. As $h_w$ becomes larger its net effect on $U_{top}$ reduces. The value of $h_w$ no longer represents the limiting factor in the heat loss and the thermal losses become more
FIGURE 33
VARIATION IN $U_{top}$ WITH $T_a$
FOR $h_w = 5$ AND 20 W/m$^2$ °C

FIGURE 34
$U_{top}$ VS MEAN ABSORBER PLATE TEMPERATURE FOR VARYING AMBIENT AIR TEMPERATURES

FIGURE 35
VARIATION IN $U_{top}$ WITH CHANGING EFFECTIVE SKY TEMPERATURE FOR $h_w = 5$ W/m$^2$ °C

FIGURE 36
VARIATION IN $U_{top}$ WITH CHANGING EFFECTIVE SKY TEMPERATURE FOR $h_w = 20$ W/m$^2$ °C
dependent on other resistances in the resistance network (see figure (8) section 4.). Specifically the value of \( h_{c,p-c} \) limits the heat transfer. Figure (32) shows the effects of varying tilt and subsequently \( h_{c,p-c} \) for two ranges of \( h_w \). At high values of \( h_w \) it may be seen that varying collector tilt more greatly affects the value of \( U_{top} \).

It may be concluded that while \( h_w \) and \( h_{c,p-c} \) both significantly affect \( U_{top} \), the effect of tilt will be more significant at higher wind velocities.

As stated, the top heat loss rate is also dependent on radiation heat losses between the absorber plate and cover glass and the cover glass and sky. This is illustrated by figure (33) where values of \( U_{top} \) are compared at different values of ambient air temperature, \( T_a \). In this situation to achieve the same value of \( (T_{pm} - T_a) \) at higher \( T_a \), the value of \( T_{pm} \) must be increased equivalently. As this is done the radiation losses, which are dependent on the absolute absorber plate temperature, increase until a point where the heat transfer becomes dominated by \( h_w \), figure (34). As \( h_w \) is increased the temperature profile through the collector increases and \( T_c \) reduces, consequently reducing the effect of \( h_{c,p-c} \) which reduces the overall effect of the increased radiative heat transfer. The situation illustrated above occurs when experimental measurements are taken in the summer when the inlet fluid temperature must be increased to achieve the same \( (T_{pm} - T_a) \) as winter.

To this point it has been assumed that the cover glass radiates energy to the atmosphere at a temperature equivalent to \( T_a \). As described in section 4.2.3.1 this is not the case for clear days. The effective sky
FIGURE 37
VARIATION OF $U_{top}$ WITH $T_a$
AND CALCULATED EFFECTIVE SKY TEMPERATURES

FIGURE 38
VARIATION OF $U_{top}$ WITH $T_a$
FOR CALCULATED $T_s$ AND
$h_w = 20 \text{ W/m}^2 \text{ °C}$

FIGURE 39
EFFECT OF $h_w$ ON $U_{top}$ ACCOUNTING
FOR REDUCED SKY TEMPERATURE

FIGURE 40
EFFECT OF COLLECTOR SLOPE ON $U_{top}$ FOR CALCULATED VALUES OF $T_s$
temperature may be significantly lower than \( T_a \). The value of \( T_s \) depends on the air temperature \( T_a \), and the portion of the sky that the collector sees.

At high collector tilt angles, the collector "sees" the ground. The ground is assumed to be at a temperature equal to \( T_a \), with an emissivity to thermal radiation of 0.9. The effect of reduced \( T_s \) relative to \( T_a \) for values of \( h_w = 5 \) and 20 W/m²°C is shown in figures 34 and 35. At low values of \( h_w \) and \( (T_{pm} - T_a) \), lower \( T_s \) significantly increases the value of \( U_{top} \). This effect is reduced at higher values of \( h_w \) and \( (T_{pm} - T_a) \), where \( h_w \) is less significant in the overall heat transfer. The effects of depressed \( T_s \) may be somewhat moderated by the fact that during experimental testing of collectors in the higher latitudes, the collector sees a large portion of the ground adjacent to it.

The program COLSM utilizes the analysis of section 4.2.3.1 to determine an effective sky temperature, \( T_s \) for the portions of the sky and ground seen by the collector. Figures (37) through (40) present the results including the effect of \( T_s \) as determined by eqn (42) section 4. At low values of \( h_w \) corresponding to low wind speed it may be seen that radiative losses to the sky are significant. This significance is reduced at higher \( (T_{pm} - T_a) \), and higher wind velocities.

6.1.2 Effective Transmittance X Absorptance Product

The value of the effective, transmittance x absorptance product is dependent on meteorological factors. The analysis leading to the calculation of \((\tau \alpha)_e\) is presented in section 4.3. The optical performance
of the glazing system of the solar collector is dependent on the incident angle of the beam solar radiation, $\theta$, and the fraction of the total incident solar radiation that is diffuse. The diffuse radiation is made up of two components; diffuse radiation reflected from the ground and diffuse radiation emanating from reflection and scattering in the sky.

The model of diffuse radiation presented in section 4.3 and implemented by the computer program OPTCAL, (described in section 5.), assumes that the effective incident angle of the two diffuse components is separate and a function of the collector slope. The slope alters the effective diffuse incident angle, $\theta_{e, dg}$, but also determines the percentage of longwave radiation received from the sky and ground. The effect of effective incident angle for ground radiation is limited by the fact that when high values of $\theta_{e, dg}$ occur, the collector is at low slope and consequently does not "see" much of the ground. The net effect is that the maximum effect of high ground reflectance occurs at slopes near $45^\circ$ and is limited to a few percentage points.

The model for diffuse radiation presented in section 4 assumes that diffuse radiation is isotropic, that is, uniform across the sky. While this is a good approximation, its validity reduces for very clear skies where the diffuse radiation may be sun centered, i.e., located near the sun disk. The overall error in this assumption for testing purposes should be small because during clear skies the fraction of diffuse radiation is low, i.e., $f_d < 20\%$. Figure 41 presents the value of $(\tau_a)_e$ calculated for ranges of $f_d$ and varying beam radiation, incident angle $0^\circ < \theta < 90^\circ$. An incident angle of zero corresponds to a direction normal to the cover surface, while $90^\circ$,
corresponds to a direction parallel to the cover surface.

![Graph showing the effect of incident angle on the value of (\(\tau_\theta\)_e) for ranges of diffuse radiation.](image)

**Figure 41**

EFFECT OF INCIDENT ANGLE ON THE VALUE OF (\(\tau_\theta\)_e) FOR RANGES OF DIFFUSE RADIATION

The net effect of increasing diffuse fraction is to lower the value of (\(\tau_\theta\)_e) at incident angles less than approximately 60°. At \(\theta > 60°\), it may be seen that increased diffuse radiation increases the value of (\(\tau_\theta\)_e). This occurs from the assumption of isotropic diffuse radiation. It is apparent that increased incident angles greater than 60° reduce the value of (\(\tau_\theta\)_e).

At incident angles less than 30° the effects of \(\theta\) are negligible. This result does not apply strictly to other collector types, as collectors with multiple glazings show more severe effects of \(\theta\) (1).
6.1.3 **Heat Removal Factor, $F_R$**

The derivation of the heat removal factor for conventional flat-plate solar collectors is presented in section 4.4. $F_R$ is a function of the fin efficiency of the absorber plate, collector efficiency factor, $F'$, and $U_L$ the heat loss coefficient. $F_R$ is strongly dependent on the collector flow rate. Low flow rates result in an increased $T_{pm}$, and lower $h_f$ and $F'$.

As flow rate increases the rate of increase of $F_R$ decreases and the value of $F_R$ approaches the value of $F'$. This situation corresponds to having the fluid in the absorber plate all at a temperature equivalent to the inlet fluid temperature. The value of $F_R$ is plotted as a function of heat transfer fluid flow rate in figure 42, for the Chamberlain solar collector. The range of flow rate obtained during the experimental portion of this project is shown for reference.

Figure 43 presents plots of $F_R$ vs flow rate for three ranges of $U_L$. It is apparent that the sensitivity to flow rate increases as $U_L$ increases. Generalizing, it may be concluded that solar collectors with high heat loss, (i.e., non-selective absorber, single glazed) are more sensitive to flow rate changes than collectors with low $U_L$, (i.e., double glazed or vacuum tube type). It may be concluded from the results of section 6.1.1 that as $U_{top}$ (and $U_L$) change for varying meteorological conditions so will the value of $F_R$. It can be concluded from figure 43, that to achieve a high value of $F_R$ in a low loss collector may require a less efficient absorber plate than a high loss collector.
Figure 42
Effect of collector flow rate on $F_R$

$U_L = 0.99$

Figure 43
Effect of $U_L$ on heat removal factor
6.2 Effect of Meteorological and Operational Factors on Collector Thermal Performance

The thermal performance of the Chamberlain collector as simulated by the COLSM program and presented in a plot of Collection Efficiency vs \((T_{fi} - T_a)/I\), is shown in figure 44. The conditions for test represent those typical for thermal performance tests, e.g.,

\[
T_a = 10^\circ C
\]
\[
\theta = 0
\]
\[
I = 1000 \text{ W/m}^2
\]
\[
S = 45^\circ
\]
\[
h_w = 15 \text{ W/m}^2\text{C}
\]
\[
f_d = 0.2
\]
\[
p = 0.2
\]

The value of effective sky temperature, \(T_s\) was calculated by the methods in section 4.2.3.1 for an inclined surface.

The results of figure 44 show that the result is slightly second order and curves downward with increasing \((T_{fi} - T_a)/I\).

6.2.1 Effects of Collector Slope

The results of section 6.1 show that collector slope significantly affects, \(U_{top}\). This result is consistent with the results shown in figure 45, where theoretical efficiency is shown as a function of collector slope. The results indicate that the large variations occur between tilts ranging from 60 to 75°. Slope effects are heat loss dependent at large values of \((T_{fi} - T_a)/I\) where convective heat loss is dependent on collector slope. The effects are much less evident at low \((T_{fi} - T_a)/I\) where convection heat loss is lower. At \(T_{fi} - T_a\) heat loss does occur because \(T_p\) is higher than
FIGURE 44
COMPUTER SIMULATED THERMAL PERFORMANCE
OF CHAMBERLAIN FLAT-PLATE SOLAR COLLECTOR

FIGURE 45
EFFECT OF COLLECTOR TILT ON THERMAL
PERFORMANCE
t and subsequently the change in \( U_L \) that occurs affects the value of \( F_R \) as described in Section 6.1.3. This effect is small. A less significant factor is that \( T_s \) increases as the collector slope increases and views the ground (-11°C @ 0° and -3.4°C @ 75°). This factor counteracts the effects of convective heat transfer slightly.

6.2.2 Effects of Wind Heat Transfer Coefficient

Figure 46 illustrates the effects of varying \( h_w \). Values of \( h_w \) equal to 5 W/m²°C represent zero wind (i.e., natural convection), while values greater than 20 W/m²°C are representative of high wind speed conditions. The effects of \( h_w \) are greater than those of collector slope. The greatest sensitivity to variations in wind speed occur at low values of \( h_w \) with ever decreasing effect at high \( h_w \). The effects of \( h_w \) are greatest at high heat loss conditions.

6.2.3 Effects of Beam Incident Angle

Figure 47 presents results for increasing \( \theta \). The results are considerable and predictable from the results of section 6.1.2 with minimal effects at \( \theta < 45° \) and great effects from \( \theta = 45° \) to 75°. During solar collector testing incident angles are limited to 30° so effects are minimized. The results presented are for a value of \( I = 1000 \) W/m² and \( f_d = 0.2 \). 

6.2.4 Effects of Insolation Level

The effects of varying insolation level, \( I \), are demonstrated in
FIGURE 46
EFFECT OF $h_w$ ON COLLECTOR THERMAL PERFORMANCE

FIGURE 47
EFFECT OF BEAM INCIDENT ANGLE
figure 48, for values from I equal to 250 to 1250 W/m². The incident angle is assumed equal to zero in all cases but the ratio of diffuse radiation is assumed to be $f_d = 0.2$. This situation does not occur in normal outdoor operation as at levels of 250 W/m² and $\theta = 0$ the sky will be overcast and $f_d = 1$. This more realistic condition is shown in figure 49. The plot shown in figure 47 for varying beam incident angle is also not representative of actual outdoor operation. Beam radiation intensity actually drops as the cosine of $\theta$. Therefore for $I = 1200$ W/m² and $f_d = 0.1$ at $\theta = 0$, the corresponding values at varying $\theta$ are:

<table>
<thead>
<tr>
<th>$\theta$</th>
<th>$I$</th>
<th>$f_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1200</td>
<td>10%</td>
</tr>
<tr>
<td>60°</td>
<td>720</td>
<td>33%</td>
</tr>
<tr>
<td>75°</td>
<td>490</td>
<td>50%</td>
</tr>
</tbody>
</table>

Figure 50 presents these results which correspond to actual outdoor operation. Luckily the experimental derivation of "incident angle modifier" as described in reference 4 should account for these effects. The results shown in figure 47 and 48 could only be achieved in a solar simulator.

6.2.5 Effect of Ambient Air Temperature

Figure 51 presents the effects of varying $T_a$. This situation occurs during outdoor testing or operation of real flat-plate solar collectors. The results are consistent with those observed in figure 33, section 6.1. As shown in figure 33 this effect will be slightly greater at lower values of $h_w$. 


**Figure 48**

Effect of Solar Radiation Level

**Figure 49**

Effect of Level of Insolation and $f_d$
6.2.6 Effect of Ground Reflectance

Figure 52 shows the effect of ground reflectance on collector performance. As discussed in section 6.1 and within the scope of the model, the effect is small even when shown for the case of \( f_d = 50\% \) and \( h_w = 5 \text{ W/m}^2\text{°C} \) as in figure 52. A greater effect of \( g \) would be due to the increased insulation level on the collector surface and the increase in \( f_d \) that results from high ground reflectance.

6.2.7 Effects of Diffuse Radiation Fraction

The effects of diffuse radiation have already been implied in the figures 49 and 50 and are shown in figure 53 independent of \( \theta \) and \( I \). The results correspond to what would be predicted in figure 41, section 6.1. The condition shown rarely occurs outdoors, (i.e., \( I = 1000 \text{ W/m}^2 \) and \( f_d = 100\% \)), but may be representative of very hazy bright winter days.

6.2.8 Effect of Operational Conditions

The effect of heat transfer fluid flow rate is presented in figure 54 for flow rates ranging from 1/10 to 10X the value recommended in (4) for collector testing. The results indicate that flowrate significantly affects solar collector thermal performance.

Figure 55 shows the effect of using water as the heat transfer fluid rather than the 50/50% mixture of ethylene glycol as used in the previous examples. The effect is of the order of a few percentage points and can be accounted to the lower specific heat of the mixture of glycol that results
**Figure 50**

Effects of varying incident angle, I and \( f_d \)

**Figure 51**

Effect of ambient air temperature
FIGURE 52
EFFECT OF GROUND REFLECTANCE

FIGURE 53
EFFECT OF INCREASING DIFFUSE SOLAR RADIATION
FIGURE 54
EFFECT OF HEAT TRANSFER FLOW RATE

FIGURE 55
EFFECT OF ETHYLENE GLYCOL OR WATER AS THE HEAT TRANSFER FLUID
in a higher $T_{pm}$ for the same insolation input.

6.2.9 Operational and Meteorological Extremes

Figure 56 presents the effects on thermal performance of extreme meteorological and operational factors. At $(T_{fi} - T_a)/I = 0$ the net effect is a difference of 6.2 efficiency points. This range increases to a value of 30.1 efficiency points at $(T_{fi} - T_a)/I = 0.1$. The range of flow rates represents values from 1 to 2X the recommended test flow rate (4) although the result would be very close for a range from 1 to 1X the recommended value. These conditions are achievable during thermal performance testing or operation during summer and winter.

6.2.10 Meteorological Extremes

If the effects of flow rate are removed from figure 56 the result is shown as in figure 57.
**FIGURE 56**

**MAXIMUM EXTREMES IN EFFICIENCY FEASIBLE UNDER EXTREME METEOROLOGICAL AND OPERATIONAL FACTORS DURING TESTING**

**FIGURE 57**

**MAXIMUM EXTREMES DUE TO METEOROLOGICAL CONDITIONS ONLY**
7. EXPERIMENTAL MEASUREMENTS

7.1 Experimental Objectives

The experimental objectives of this project are summarized below:

1. To construct a test apparatus to determine the thermal performance of a typical flat plate solar collector and to monitor the meteorological factors experienced during testing.

2. To measure the thermal performance of a well characterized flat plate solar collector to determine the effect of meteorological factors on the thermal performance test results, independent of test sample variability and systematic errors.

Data should be recorded for a variety of atmospheric conditions which are representative of the range of values that occur during testing performed in the Canadian climate. Only data that meets the requirements of the ASHRAE 93-77 test method (4), should be considered.

7.2 Solar Collector Test Facility

An outdoor test facility was constructed on the roof of building M-24 of the Montreal Road laboratories of the National Research Council to determine the thermal performance of liquid based solar collectors. Test samples are mounted on a fixed frame facing due south and provision was made to adjust the tilt angle of the test collector. Tilt angles ranging from 20° to 70° relative to horizontal may be achieved to maintain the surface of the solar collectors perpendicular to the beam solar radiation at solar noon, (i.e., normal incidence). Heat transfer fluid is pumped to the solar
collectors under test from an equipment room located directly below the test racks.

A schematic drawing of the fluid test loop is shown in figure (58). It consists of a temperature control liquid flow loop and a calorimetry, (i.e., heat-flow measurement), circulation loop. The equipment and instrumentation were designed and selected to conform to the general requirements of the ASHRAE testing procedure (4). The facility has two calorimetry flow loops draining into two interconnected storage tanks of approximately 150 litres total fluid capacity. The heat transfer fluid used for the tests was a 50/50% mixture of ethylene-glycol and water to enable testing to be performed year round.

The temperature control loop circulates liquid from the storage tanks through an electric heater and chilled water heat exchanger to maintain a fixed temperature in the storage tanks. Three power levels are possible on the heater to obtain the best control. Fine control of the tank temperature is accomplished by controlling the flow of chilled water through the heat exchanger. This is accomplished by the use of an electronic proportional control that senses the storage temperature and actuates a proportional pneumatic valve on the chilled water supply to the heat exchanger.

A centrifugal pump circulates the heat transfer fluid from the storage tanks through the temperature control loop at approximately 45 litres per minute. The net result is that the storage tanks effectively act as a well mixed constant temperature bath from which the calorimetry test loops draw. The system as constructed maintains a fixed bath temperature within ±0.05°C/hr and typically ±0.1°C per day. Inlet temperature
FIGURE 58 SCHEMATIC OF DBR/NRC SOLAR CALORIMETER APPARATUS
fluctuations are usually less than 0.02°C over a test interval, (approximately 15 min.).

The heat transfer fluid is drawn (at constant temperature) from the storage tanks by a centrifugal pump and pumped to the roof for circulation through the test solar collector in a closed loop. Flow rate control through the test loop is performed by throttling the flow with a needle valve installed downstream of the pump. This worked well and maintained flow rate effectively constant at a set point for a fixed fluid temperature, but the flow rate did change when the fluid temperature was changed. This was due to the changes in fluid properties with temperature and the subsequent change in the pressure drop through the test loop. This was not a major problem but required that the flow be readjusted after a change in fluid temperature. This condition was later reduced by replacing the needle valve with a Cuno cartridge filter and Kates Automatic Flow Rate Controller. The flow rate through the calorimetry test loops could be set at values ranging from approximately 0.5 to 5 litres per minute.

The flow rate through the test collector was not measured by the method described in the ASHRAE test method (4), which consists of measuring the volume flow rate and multiplying this by the density of the fluid to determine the mass flow rate.

The thermal performance of solar collectors is determined over a wide range of heat transfer fluid temperatures (4), and the thermal properties of the heat transfer fluid are a function of the fluid temperature. Often the physical properties of heat transfer fluids (i.e., glycol-water mixtures) are poorly characterized and the accurate determination of collector thermal
performance requires the measurement of the product of flow rate and absolute specific heat, \( \text{C}_p \). Volumetric flow meters must be calibrated to account for variations in fluid density and viscosity with temperature. Considering the duration and accuracy requirements of this experiment, and evidence of the drifting of volumetric flowmeter calibrations with time (Jenkins)(66); a direct comparison reference heat source (RHS) was utilized in the calorimetry loop to determine the rate of energy collection of the test specimen. This technique has been referred to as the "Calorimetric Ratio Technique" as described by Reed and Allen (67). The reference heater is installed in series with the collector under test. Since the mass flow rate through the reference heater and the test solar collector are the same the following holds:

\[
\frac{Q_u}{\text{C}_p \cdot \Delta T} = \frac{Q_{\text{RHS}}}{\text{C}_{\text{RHS}} \cdot \Delta T_{\text{RHS}}}
\]

where

\( Q_{\text{RHS}} \)

is the rate of energy input to the fluid as it passes through the RHS.

\( \text{C}_p, \text{C}_{\text{RHS}} \)

are the specific heat of the heat transfer fluid in the solar collector and reference heater, respectively.

\( \Delta T, \Delta T_{\text{RHS}} \)

are the temperature rise in the heat transfer fluid as it passes through the solar collector and reference heater respectively.

Equation (4-1) gives:

\[
Q_u = Q_{\text{RHS}} \cdot \frac{\text{C}_p \cdot \Delta T}{\text{C}_{\text{RHS}} \cdot \Delta T_{\text{RHS}}}
\]
For most test conditions, the difference in fluid temperature between
the reference heat source and the test collector can be kept small and the
difference in fluid specific heat at the heater and collector can be
neglected. The value of $Q_{RHS}$ is equivalent to the electrical energy input
to the reference heat source minus any heat losses from the reference heater
to the surroundings. Thus:

$$Q_{RHS} = P_{RHS} - QL_{RHS}$$

(101)

The value of $P_{RHS}$ was monitored continuously by an Scientific Columbus Power
Transducer. The value of $QL_{RHS}$ was determined by measuring the heat loss
across the heater at a series of mean fluid temperatures above and below the
ambient air temperature surrounding the RHS, and applying this correction to
the value of $P_{RHS}$.

The piping of the calorimeter test apparatus was insulated to
increase the temperature stability and reduce the temperature difference
between the reference heater and solar collector under test. A 750 watt
charridge type heater was used for the reference heater. The calorimetric
to technique like conventional solar collector testing is subject to
errors due to non-steady conditions. The reference heat source has thermal
mass and, as such, can store and release energy during transient operating
conditions resulting in an over-or under-prediction of collector
performance. For this reason, flow rate, inlet temperature, heater power,
and subsequent temperature rise must be held constant during each test
period. Reference (4) outlines requirements for steady state testing of
solar collectors and these conditions are consistent with those required by the reference heat source. The power input to the reference heater was maintained at a constant value by a Stabiline automatic voltage regulator and the inlet temperature to the RHS was held at fixed temperature (as described) by the temperature control loop.

7.2.1 Temperature Measurements

Temperature difference measurements across the reference heater and the solar collector were made utilizing 8 junction, type T (copper-constantan) differential thermopiles. These devices were reliable and provided a high degree of resolution for the reasonably small temperature rise experienced during testing. Absolute fluid temperature measurements were also made at the inlet and outlet of both the RHS and the test collector using Type "T" thermocouples imbedded in each end of the thermopile devices. Both the thermocouple and the thermopiles were calibrated by the Standards Division of the Physics Division of NRC against primary standards. The accuracy of each of the calorimetric measurements and the subsequent experimental uncertainty is presented in Appendix H. The temperature sensors were inserted in insulated temperature wells inserted parallel to the fluid flow. Fluid mixers were utilized prior to each temperature measurement. Temperature measurements were made as close as possible to the inlet and outlet of both the test collector and the reference heater and the connections insulated to minimize any heat losses.
7.2.2 Meteorological Measurements

The specifications of the meteorological instrumentation are given in Appendix E. Absolute ambient air temperature was measured with a calibrated type "T" thermocouple situated in a well ventilated "Stephenson" screen enclosure installed approximately 1.3 m above the roof of the laboratory adjacent to the test frame. The shelter was installed with the door facing south to ensure that illumination by direct solar radiation could not occur. The wind velocity was measured by a three cup wind anemometer delivering a dc voltage proportional to wind velocity. A directional vane was also incorporated in the device to indicate prevailing wind direction. The wind velocity measurement was located at a point mid height to the collector under test and located approximately 3 meters due south of the test frame. The device was located so as to not shade the collector under test but it was found that the wind speed and direction were highly variable in this location. This is attributed to the fact that the test rack was located approximately 4 meters from the corner of the roof of the building. The wind flow in this location was influenced by the geometry of the building. It was discovered that more consistent data could be achieved by using the wind speed and direction data recorded from a wind anemometer and directional vane located on the top of a 7.6 meter tower located on an adjacent laboratory building located approximately 100 meters to the south of the test facility. It is considered that, while the wind velocity indicated by device was not indicative of the instantaneous local condition at the test collector, it did represent the mean free stream condition of the prevailing wind. Mean values obtained from this equipment are representative of the average meteorological conditions.
This wind velocity and direction measurement is made by the Atmospheric Environment Service for the NRC location and mean hourly values of wind speed and direction were combined with the measured collector performance data from AES meteorological summaries for the test periods.

A "black-and-white" Eppley model 8-48 pyranometer was used to measure total solar radiation incident on the collector surface. The device was mounted facing due south, tilted at the same tilt as the collector under test. The pyranometer was positioned to minimize reflected and re-radiated energy from the solar collector on to the pyranometer.

Total horizontal solar radiation was measured by a model 10 Eppley pyranometer, mounted in a horizontal plane and located where it would not be shaded by surrounding structures. The diffuse component of the incident solar radiation was measured continuously by use of a Shade Ring Diffusograph (68), in conjunction with a model 50 Eppley pyranometer mounted horizontally.

7.2.3 Data Acquisition

The specifications of the data acquisition system are given in Appendix E. The data from the calorimetric measurements (i.e., temperature differences, absolute temperatures, heater power) as well as meteorological data (ambient air temperature, total solar radiation incident on collector, total horizontal radiation, diffuse radiation) were monitored continuously during the test intervals and recorded by a central data acquisition system. As discussed previously, wind speed and direction were merged with the data during data processing. The data system was located adjacent to the
calorimeter equipment located directly below the test frames. The data
system used was a 60 channel KAYE Model 8000 data logger. Input signals
included analog voltage from the pyranometers, thermopile, thermocouple and
power transducers. Of these the analog signals from the thermocouples are
converted by the data logger to temperature values utilizing an automatic
electronic reference junction. The resultant analog voltage and temperature
readings were recorded on 9 track 800 BPI, magnetic tape using a Cipher tape
unit.

Instantaneous readings of all the calorimetry and meteorological
factors were recorded at an interval of once per minute during the testing.
The interval is considered to be fast enough, due to the steady state nature
of the experiment, and the fact that, as required by the ASHRAE test method,
testing was only carried out on clear sunny days. The value of total solar
radiation was recorded continuously on a strip chart recorder to ensure
that the requirement for steady solar radiation conditions was met during
testing.

The data recorded on magnetic tape was then processed and combined
as discussed in Section 7.4, utilizing the computer system described in
section 5.

7.3 Experimental Procedure

Tests were conducted during a period spanning mid-summer through the
winter to spring. Data was recorded every day regardless of the weather
conditions. On clear sunny days that met the basic requirements of (4), the
following steps were taken in preparation of the tests.
1. The collector was checked to ensure that it was not damaged or had ingressed moisture into its housing.

2. The meteorological measurement equipment was checked and any precipitation, snow, or ice was removed. Radiometers were wiped with water or alcohol as the ambient temperature dictated.

3. The shadow band was checked and adjusted as required.

4. The collector inlet fluid temperature was held at the set point temperature during the day and set at a new value in the evening.

The tilt of the collector was adjusted periodically to maintain the incident angle to beam radiation within the ASHRAE requirements (4). Particular attention was paid to the accumulation of moisture in the collector casing. Moisture accumulation occurs as a result of the collector "breathing" during the day. Air drawn into the collector contains moisture which condenses on the cold glass surface during the night. This moisture drains to the bottom of the collector where it is absorbed by the fiberglass insulation in the collector. During the day this water evaporates and condenses on the colder glass surface in a continuous stream. In a very short time, if precautions are not taken the insulation becomes saturated, severely degrading the collector thermal performance.

Most solar collectors are equipped with drain holes at their lower edge to drain off water accumulation. The Chamberlain collector did not use this system but was equipped with a self re-generating desiccant system intended to limit moisture accumulation. The Canadian climate seemed too much for this system and it could not regenerate sufficiently. To
counteract moisture accumulation, the collector was purged with dry air every night. The operation was controlled by a timer and shut off early in the morning so as not to affect the tests.

This process of purging ensured that the collector heat loss characteristics remained fixed and a major source of error in the experiment was reduced.

7.4 Data Processing

At the end of a test period consisting of approximately three and a half days, the full data tape was removed from the Cipher tape unit and replaced with a blank tape. These tapes were stored for processing.

Raw data tapes from the data logger contained digital records of the collector inlet and outlet temperatures, ambient air temperature, horizontal total and diffuse solar radiation, total solar radiation on the collector slope, and the millivolt reading of the differential thermopile output, across the collector inlet and outlet. The input power to the reference heater was also recorded on the tape.

This raw data stored in one minute records also contained a record with the date, (year and julian day of the year, i.e., 1-365), and time for reference.

Data processing was accomplished in the following manner;

1. Data tapes were processed on a monthly basis by computer.

2. Each three day tape was scanned to select only data from between the hours of 10:00 AM to 2:00 PM EST, to ensure that incident angles were low.

3. The calorimeter test data was merged with average wind speed and
direction for the corresponding hour.

4. Each data set was scanned during the test period (i.e., 10:00 AM to 2:00 PM), to locate periods of steady state that met the criteria of (4). Steady state conditions were defined as a period of time equal to 5 collector time constants long, (as measured in (4)), and meeting the following requirements:

(i) Solar radiation in collector equal to the average over the period ± 20 W/m²
(ii) Collector inlet, outlet and ambient air temperature equal to average over period ± 0.1 °C
(iii) Temperature rise in the collector equal to average values over period ± 0.05 °C

5. Only data that met the steady state criteria was selected by the computer and stored on magnetic tape.

6. Data records of "steady" data were stored on "monthly", magnetic tapes.

7. These "monthly" data tapes were then combined into a "yearly" data tape containing only data that met the steady state requirements.

8. The "yearly" data tape was transferred onto rigid disk on the computer system described in section 5.

During the data processing, raw data was converted to engineering values by applying the appropriate conversion factors. The temperature rises across the collector and reference heater were derived from the differential thermopile by utilizing the thermopile's sensitivity at the average of the inlet and outlet temperatures of each. The rate of energy collection was determined utilizing the corrected reference heater power
input, and temperature rise and the collector temperature rise as in equation (100), assuming $C_{p_{RHS}} = C_{p_{C}}$. This assumption should typically introduce errors of less than a $\frac{1}{4}$ percent. The solar energy incident on the collector surface was determined from the pyranometer output voltage signal and the calibration factor supplied during the Atmospheric Environment Service calibration. Collection efficiency was determined by;

$$\eta = \frac{Q_u}{IA_c}$$

where the reference area for the Chamberlain collector was equal to the aperture area = 1.79 m².

Tests were conducted on the Chamberlain collector to determine its time constant as described in reference (4). The value obtained was approximately 1½ minutes for one time constant in the classical sense. A steady state interval equal to 8 minutes was chosen to process the data and subsequently 8 minute average values were produced.

The resultant data was processed and analyzed as presented in section 7 by the computer system described in section 5 utilizing a software package described in reference 61.
8. EXPERIMENTAL DATA

Data was collected from June 30, 1978, (Julian day No. 181), to May 19, 1979, (Julian day No. 139). During that time, days that had clear skies and periods that met the steady state criteria between the time slot from 10:00 AM to 2:00 PM were very limited. In all 67 days contained data that was suitable for analysis, consisting of 200 data points.

8.1 Test Data

The test data obtained from the experiment is listed in Appendix F, by file number consisting of the test circuit no., collector test slope, year and Julian day. For example, 1C2278.181 corresponds to test circuit 1, Chamberlain collector, 22° tilt, day 181, 1978.

8.2 Analysis of Data

The total data set is plotted in Figure 59 for collection efficiency vs $(T_{f1} - T_a)/I$.

To determine if a wide range of meteorological conditions had been obtained during testing the data was analyzed in regard to operational and meteorological frequency. The results of this analysis are shown in Appendix G.

The following data ranges were determined;

Windspeed: 0 to 7 m/sec,

Ambient Air Temperature: -20 to 30°C

Total Solar Insolation: 750 to 1100 W/m²

Horizontal Diffuse Solar Radiation: 50 to 200 W/m²

Incident Angle, $\Theta$: 0 to 40°

Flow rate: 1.8 to 2.6 L/min.
FIGURE 59
PLOT OF EXPERIMENTALLY DETERMINED THERMAL EFFICIENCY FOR THE CHAMBERLAIN SOLAR COLLECTOR
9. DISCUSSION—EFFECT OF METEOROLOGICAL FACTORS ON TEST RESULTS

9.1 Discussion of Theoretical Results

The theoretical effects of meteorological and operational factors, on the thermal performance of the Chamberlain solar collector have been presented in section 6.

While operational factors significantly affect the thermal performance of flat-plate solar collectors, it is assumed that for the purposes of testing collectors, variations in flow rate may be minimized.

If meteorological factors only are considered in the analysis, as shown in figure 57, the divergence of results is reduced. Collector slope is included in meteorological factors as its value is a consequence of obtaining "normal" incident angles during testing for different seasons of the year.

For the Chamberlain collector, it is apparent that wind speed, $h_w$, collector slope, ambient air temperature and fraction of diffuse solar radiation significantly affect collector thermal performance. It is interesting to note that the effects of these parameters are all to cause a reduction in the maximum thermal performance achievable in a rating test. That is to say, if an "informed" manufacturer chooses when and where he has his product tested and specifies what he will accept for meteorological factors, he may obtain the maximum values predicted in figure 57. This in fact may not represent the actual performance of the collector in an operating system. Consequently, there is no "margin of safety" designed into the system performance evaluation as in common design practice. A considerable overprediction of system performance may occur.
If the results of section 6, for the figure 57, are re-run, limiting the meteorological extremes allowed under the standard thermal performance testing document (4), the result is shown in figure 60.

Figure 60 indicates that at $T_{fi} - T_{fi}$ the deviation in $\eta$ is reduced to a couple of percentage points. At $(T_{fi} - T_{fi})/I$ equal to 0.1 the variation is 23 percentage points. The output of a collector at the low range would be half that of the maximum at this operation point. It should be noted that in actual use collectors operate at this end of the performance curve for much of the time.

Reviewing the results of section 6.2 it may be determined that the range of values achievable is largely due to the lack of restriction on collector slope and $h_w$. Wind speed allowed in (4) ranges from 0 to 4.5 m/sec, corresponding to $h_w = 5$ to approximately 30. The ranges of collector slope shown in figure 57 are from 20° to 70°, which is typical for summer and winter extremes in Canada. If a manufacturer was to have his product tested at lower latitudes, (i.e., 30°N Lat.), representing the southern USA, collector slopes of 0° are possible and the effect would be even greater.

The range of wind velocity specified in (4) represents the range of greatest sensitivity to $h_w$, and therefore is subject to the maximum changes in $\eta$ for a change in $h_w$. This not only results in scatter in the test data but also may bias the performance curve to the extremes shown in figure 57. It has been suggested (44), that for simulator collector testing, the minimum wind speed be increased to that corresponding to a value in the range 2.25 to 4.5 m/sec. This would correspond to the mean value in Canada and reduce scatter in the test result. The same recommendations can be made
FIGURE 60
EFFECT OF METEOROLOGICAL CONDITIONS
ALLOWED WITHIN TEST PROCEDURE (4)

FIGURE 61
EFFECT OF LIMITING METEOROLOGICAL
CONDITIONS DURING TESTING
for outdoor testing. Figure 61 demonstrates this fact, showing that if $h_w$ is held to a range corresponding to 20 to 30 W/m$^2\cdot$C, and limiting the collector tilt to 45°, the result is to reduce the extreme range to 7.5 percentage points. This would "weight" the test result to the most common operational value of $h_w$. Increasing the upper limit of $h_w$ to a value above 30 W/m$^2\cdot$C has little effect on the results as $h_w$ is no longer a limiting factor in the heat transfer. Specifying a fixed collector slope of 45° will cause the requirement of (4) that $\Theta > 30^\circ$ to still be met at this latitude. The slope of 45° is typical of solar installations in Canada. The increased restrictions on $h_w$ should not significantly increase the time required for a specific test as the highest frequencies of wind speed occur near these values. Still air conditions rarely occur as described in Section 8. The fact that low wind speeds are allowed under the present standard encourages the use of specific data points with low $h_w$ to generate a higher performance curve.

A further reduction in the requirements for $f_d$ and $\Theta$ may achieve better results than Figure 61 but at the cost of increased testing time. The requirement for steady state conditions described in (4) virtually eliminates radiation levels between 200 to 900 W/m$^2$ because the atmospheric conditions are too unstable to produce steady state values. Increasing the minimum allowable insolation level to 900 W/m$^2$ should have no effect on the required testing time for outdoor tests.

This value is greater than that recommended (16) as representing an average value for Canada of $I = 650$ W/m$^2$. The increased $I$ causes a small reduction in the thermal performance predicted, but the effect will be conservative from the designer's point of view. Periods when insolation
rate is very low, i.e., 250 W/m², represent times when \( n \) is very low and consequently, represent a small portion of the total collected energy. This "weighting" the test to higher radiation levels does not seem unreasonable to achieve steady test conditions.

9.1.1 Significance of the Results for Other Collector Types

The results can be applied to other single-glazed flat plate collectors with selective absorber surfaces, without significant error. The Chamberlain represents a high performing collector, largely due to its good selective surface (i.e., \( \varepsilon_p = 0.95, \varepsilon_p = 0.12 \)) and its very effective absorber plate (\( F = 0.99 \)). The effect of the selective surface is to reduce the value of \( U_L \) by reduction of radiation losses, \( h_{r,p-c} \) and \( h_{r,p-s} \) and to subsequently lower the glass temperature, reducing the effect of \( h_w \). A single glazed collector with non-selective absorber, (i.e., \( \alpha_p = \varepsilon_p = 0.95 \)), will increase the effect of all meteorological factors. As an example the COLSM simulation was run for the case of \( \varepsilon_p = 0.95 \) and the same meteorological factors as presented in figure 60. The effect of increasing \( \varepsilon_p \), is to increase the spread between the extremes due to meteorological factors, figure 62.

If the test conditions were limited as described for figure 61, the result is shown in figure 63 for the non-selective case. The maximum deviation is also decreased. If the glazing was not glass, which has a low transmittance to longwave radiation, and was for example "Teflon" plastic the effect of meteorological factor would be as shown in figure 64. "Teflon" film has a transmittance to longwave radiation of 0.32, (62). For
FIGURE 62
EFFECT OF $\epsilon_p$ ON METEOROLOGICAL EXTREMES

FIGURE 63
EFFECT OF LIMITING METEOROLOGICAL EXTREMES FOR NON-SELECTIVE ABSORBER
this case the collector is more dependent on effective sky temperature. The range of values is actually reduced but with an associated drop in thermal efficiency. The maximum effect of longwave transmittance to the sky is shown in figure 65, to indicate a worst case for the "Teflon" cover. The effect is significant.

The COILSM routine does not presently have the capability to simulate multiple glazed collectors but a double glazed, non-selective absorber plate, collector should respond similarly to the single glazed, selective absorber plate case as the values of $U_L$ are similar (1). The effect of $T_s$ should be very small and the performance curve will be uniformly reduced indicating the loss in transmittance due to the extra sheet of glass in the cover system.

Unglazed collectors similar to those used for swimming pools will be very sensitive to all the meteorological factors. Luckily, their operation is usually limited to low fluid temperatures relative to air temperature, i.e., swimming pool temperature. The dependence of $F_R$ on $U_L$ dictate that the fluid:flow rate must be kept very high to limit $T_{pm}$. A study to determine the sensitivity of $F_R$ as a function of flow rate on $h_w$ is an area recommended for future study.

Double glazed, selective absorber surface, collectors and vacuum tube collectors (40) are characteristic of low $U_L$ and subsequently should be less sensitive to meteorological factors than the Chamberlain collector. The reduced convection losses between the absorber and cover in a vacuum tube type collector seem to indicate that the governing heat losses will be by radiative exchange between the absorber and cover and subsequently be
Figure 64
EFFECT OF $\varepsilon_p$ AND $T_{LW}$ ON METEOROLOGICAL EXTREMES

Figure 65
MAXIMUM EFFECT OF $T_s$ FOR $\varepsilon_p = 0.95$ AND $T_{LW} = 0.32$
dependent on \( h_w \) and \( T_{pm} \). At low fluid temperature, i.e., \( T_f \), < 100°C, losses should be small but this type of collector is usually specified for higher temperature applications such as industrial process heat etc. The effect of meteorological factors may be significant in these applications. These effects require investigation.

The results presented for the Chamberlain collector are representative of the greatest portion of commercially available collectors. For this reason the results should be applicable to a broad range of applications.

9.1.2 Significance to Simulator Testing

Solar collector testing indoors using an artificial source of solar radiation has been suggested as one means of eliminating variation in test results due to meteorological conditions. Solar simulators have been described (63, 64) and while offering the capability to standardize the test condition, the relationship between simulator tests and outdoor operation must be determined. Solar simulators are characterized by all beam solar radiation, i.e., \( f_d = 0 \) and significant amounts of longwave radiation produced by the hotter than normal atmosphere. Simulator lamps have been known to produce 100 W/m² of longwave thermal energy above what would occur in the ambient environment.

It has been shown that the effects of effective sky temperature may be reduced by increasing the value of \( h_w \) so that convective heat transfer dominates the collector heat loss. Testing collectors that have covers that are transparent to longwave may present significant problems.
The effect of total beam radiation will tend to overpredict the collector performance slightly as will the increased effective sky temperature.

The incident angle modifier test as described (4), presents problems similar to those described in section 6.2.4. The result is that incident angle effects will be more severe if measured under all beam radiation as opposed to those measured with diffuse sky components. Corrections similar to those applied in 6.2.4 should be possible to adjust simulator results to outdoor conditions.

9.2 Discussion of Experimental Results

The experimental results are presented in section 8. The data is shown plotted in figure 59. A curve fit by the method of "least squares", produces the equation:

\[ \eta (T) = 77.1 - 422 \frac{(T_f - T_o)}{I} \]

with a standard error = 5.7, (71)

This represents a value for;

\[ F_R (T_0) = 0.77 \]

\[ F_R \frac{UL}{UL} = 4.22 \]

These values fall within the values derived by the COLSM Simulation for the range of meteorological extremes, figure 57.

The experimental data shows the same trends as that of the theoretical data but the meteorological extremes are not as severe. This can
be accounted to the lack of data during the most extreme cases. This illustrates one important point that for this location the extreme conditions do not readily occur. The data is consistent with theory in that it is more scattered at operational points corresponding to high thermal loss.

The scatter that is shown, is a result of varying meteorological conditions combined with experimental error associated with performing the experiments. An analysis of experimental error associated with the experiment is presented in Appendix H. The scatter due to meteorological conditions may be determined utilizing the COLSM routine with the appropriate range of meteorological and operational factors (figure 66) experienced during the test. Experimental uncertainty alone cannot account for the scatter in the test results. Meteorological conditions experienced during the test combined with experimental uncertainty do account for the scatter in the test data. The solid lines shown in figure 66 represent the meteorological extremes and it should be noted that actual collector operation occurred between these extremes. With a large data set such as obtained during this testing, a curve fit analysis of the data should be fairly representative of the mean operational condition if the data has uniformly distributed meteorological factors. The analysis in section 8 demonstrates that all the data consisted of periods with high solar insolation levels and relatively low diffuse radiation. This is a consequence of the steady state requirement. In effect only very clear, bright days have sufficiently steady state characteristics to meet the requirements of the test procedure (4). The test result is subsequently
biased toward clear sunny days with very low diffuse radiation levels. The consequence of this is discussed in section 9.1.

While the data is fairly uniformly distributed, the standard test procedure calls for only four sets of data to be obtained, evenly distributed along the \((T_{fi}-T_{a})/I\) axis. Subsequently any four data points could be chosen for the curve fit. If the data on particular days was biased then significant differences in the predicted average performance would be obtained.

The limit in the scatter of the data, as stated, is a consequence of the limited extreme meteorological conditions experienced during testing at the Ottawa site. It should not be concluded that further extremes may not be achieved, but only that they were not attained during this experimental period.

It is common practice for solar collector manufacturers to ship their products to different test labs across North America. The difference between results obtained in largely different climates may be significant.

9.3 Comparison of Theoretical and Experimental Results

Figure 66 presents the experimentally derived values of collector thermal efficiency. The range of meteorological extremes as predicted by the COSLM computer simulation as represented by the solid lines shown on the plot.

To determine the accuracy of the COLSM program to calculate the absolute value of theoretical efficiency, direct comparisons of calculated and measured \(\eta\) were compared for identical meteorological and operational
Figure 66
Plot of experimental data showing theoretical meteorological extremes and measurement uncertainty.
conditions.

It was observed that at high efficiency values where thermal losses from the collector are minimal, that the COLSM program over-predicted experimental efficiencies by approximately 4 to 5 percentage points.

This may be attributed to a number of effects. The optical and heat transfer properties used in the program are those measured on unweathered collectors. The collector used for experimental testing was well aged before the tests as it was used in other outdoor experiments. Fouling of the inner surface of the flow passages of the absorber plate is not accounted for in the COLSM program. The absorber is constructed of carbon steel and it is possible that this may have slightly influenced the absorber plate characteristics.

Variations in performance between new and weathered solar collectors (specifically the Chamberlain solar collector), have been shown (69). The deviation between the experimental and theoretical results is close to that experienced previously (69). Reference (70) concluded that a significant drop in absorber plate absorptance occurred after weathering of air collectors, but heat loss coefficient did not change appreciably.

Assuming that the optical properties on the test collector may be slightly less than those stated, values of \( \varepsilon_p \), were adjusted in the COLSM program to produce consistent results. A value of \( \varepsilon_p = 0.90 \) was chosen finally as being representative of the weathered value of \( \varepsilon_p \). The effect of varying \( \alpha_p \) on the value of \( (\tau a)_e \) is shown in figure 67.

To compare the experimental data with the simulated values, a value of \( h_w \) must be determined for the experimental case. This parameter was
Figure 67
Effect of Absorber Plate, Absorptance on \( \alpha \)

Figure 68
Comparison of Solar Collector Thermal Efficiency as Determined by the COLSM Program and Experimentally
not measured directly in the experiment and the difficulties with predicting its value from wind speed data has been discussed in section 4. To resolve this problem, the various relationships for $h_w$, as described in section 4, were utilized to determine $h_w$ for the experimental data. Experimentally derived values of $n$ were then compared to simulated performance at the predicted $h_w$. Surprisingly, the best match of the experimental and theoretical data occurred with the relationship originally presented by Jürges, eqn. 46 section 4. The comparison of the resultant experimentally derived efficiency values are plotted against those predicted by COLSM for the same meteorological and operational conditions, figure 68. The dashed lines in figure 68 represent the average experimental uncertainty associated with the experimental measurements.

The data for $\phi_p = 0.90$, $U_e = 0.1$ W/m²°C and $h_w$ as predicted by eqn. 46 fall within the experimental uncertainty.

The success of eqn. 46 in predicting the value of $h_w$ may be due to the fact that wind speeds were relatively low for the experiments, typically less than 6 m/sec, and the fact that the location of the collectors, free standing on the roof of a building, probably resulted in a highly turbulent flow. The effects of wind direction for this situation would be minimized and this conclusion was supported by the experimental data that showed no correlation to wind direction.

9.4 Normalization of Experimental Results

Normalization of experimental data to standard experimental conditions has been suggested as one method of reducing scatter and biased
test results, (16, 71, 72). The method requires that an accurate model of
the collector heat transfer is available. The complexity of the COLSM model
for the fairly uncomplicated Chamberlain flat-plate solar collector
illustrates part of the problem with a universal application of the theory
for all collector types. Models of thermal performance will be required for
air and liquid flat plate collectors, vacuum tube type collectors, unglazed
collectors and concentrating collectors.

In normalizing theory, it is assumed that a relatively simple
collector heat transfer model can predict changes in performance due to
varying meteorological conditions better than predicting the absolute
collector efficiency at a specific meteorological condition.

Values of experimental efficiency are measured at the experimental
meteorological condition. With the meteorological conditions determined
during the experiment, values of simulated collector performance are
calculated for various values of edge heat loss coefficient, $U_e$, and absorber
plate absorptance, $\alpha_p$, until a match is obtained between the simulated and
experimental values of thermal efficiency. Values of $\alpha_p$ and $U_e$ that provide
the best fit to all the data obtained during the experiment are used for the
normalization. Thus, the computer program is "fit" to the experimental
data similar to the analysis given in section 9.3. This fitted collector
model is used to predict the performance of the test collector at a standard
meteorological condition. The difference between the experimentally
determined values of $\eta$ and those simulated by the collector model at
standard conditions are then applied to the experimental data to "correct"
it to the standard condition.
This method was applied to the test data derived in this study utilizing the values of $\alpha_p$, $U_e$, and $h_w$ determined in section 9.3.

The normalized data is presented in figure 69. Values were normalized to the condition of:

$$h_w = 17 \text{ W/m}^2 \text{ °C}$$
$$I = 650 \text{ W/m}^2$$
$$\theta = 0^\circ$$
$$T_a = 0^\circ\text{C}$$
$$\rho_g = 0.2$$
$$S = 45^\circ$$
$$f_d = 0.3$$

It may be seen from figure 69, that the scatter in the data is reduced to a value near the experimental uncertainty of the outdoor tests. Experimental uncertainty represents the limits of accuracy of the normalization procedure assuming the collector model is accurate.

The value of collector performance determined for the standard condition is:

$$n = 75.3 - 408 \frac{(T_{fi} - T_a)}{I}$$

with a standard error of 3.2

The normalized data showed a better fit to a second order curve.

$$n = 74.7 - 364 \frac{(T_{fi} - T_a)}{I} - 504 \left( \frac{(T_{fi} - T_a)}{I} \right)^2$$

with standard error equal to 3.1

This better correlation with a second order relationship is consistent with
the theoretical results that also exhibit a second order characteristic. This characteristic was not apparent in the unnormalized data.

Reviewing figure 69 and the unnormalized data it may be observed that the three data points circled in figure 69 did not move during the normalization process. This fact can be attributed to the fact that these points were at the standard meteorological condition to start with or that the data is in error. A review of the original data showed that they were in fact not taken at the standard condition and that the values of $dT_c$ were incorrect, so these points must be assumed to be "in error". If they are eliminated from the data set the resultant curve fit gives;

$$\eta = 74.8 - 622 \left( \frac{T_{fi} - T_a}{I} \right) - 528 \left( \frac{T_{fi} - T_a}{I} \right)^2$$

with a standard error = 2.6

These results show that although scatter still exists in the data the values are reduced to within the uncertainty of the experimental values.

An interesting outcome of this analysis has been to indicate errors in data that were not evident in the original data due to meteorological scatter.

An attempt was made to normalize the data to high efficiency values, figure 70. A best fit of the data was given by;

$$\eta = 75.8 - 276 \left( \frac{T_{fi} - T_a}{I} \right) - 406 \left( \frac{T_{fi} - T_a}{I} \right)^2$$

with a standard error = 2.6

The standard conditions in this example were:

$$h = 5 \text{ w/m}^2 \text{ C}$$

$$I = 1100 \text{ w/m}^2$$

$$\theta = 0$$
\[ T_a = -20^\circ C \]
\[ \rho_g = 0.7 \]
\[ S = 45^\circ \]
\[ f_d = 0.3 \]

In both examples the scatter in the data was not eliminated but has reduced to a value consistent with the experimental measurement uncertainty. The ultimate success of this procedure is dependent on the accuracy of the collector model and the measurement uncertainty of the experimental measurements. The experimental measurement uncertainty is dominated by the uncertainty of the solar radiation measurement, and the collector model is dominated by the uncertainty of the determination of \( h_w \), \( T_{sky} \), and diffuse radiation faction.

Instruments for more accurately measuring solar radiation are now available and are being improved and characterized. Instruments for measuring \( T_s \) are now available (21), but their accuracy is still unknown.

The determination of \( h_w \) from wind speed measurements seems to be a major weakness. It is suggested that equipment be developed and utilized to determine the value of \( h_w \) present during the particular test. The relationship between \( h_w \) and free stream wind velocity should be further investigated so that deviations of solar collector average performance can be made for specific applications.
FIGURE 69
PLOT OF EXPERIMENTAL DATA NORMALIZED TO
STANDARD CONDITIONS.

PLOT OF SOLAR COLLECTOR EFFICIENCY

FIGURE 70
PLOT OF DATA NORMALIZED TO STANDARD CONDITIONS
REPRESENTATIVE OF HIGH COLLECTOR PERFORMANCE
9.5 Recommended Conditions for Thermal Performance Tests

Section 9 discusses the results of the theoretical and experimental results. The range of test results achievable within the limits of meteorological factors specified in the standard test procedure (4) are indicated in figure 60. Figure 61 shows that if meteorological extremes are further limited, the range of possible test results may be considerably reduced.

The major contributors to the range of possible test results are \( h_w \) and collector slope. If values of \( h_w \) are limited to values greater than 20 W/m² and collector tilt is held near a value of 45° then consistent test results may be achieved more readily.

As described in section 9, this restriction will result in slightly conservative test results but these would be more representative of average meteorological conditions experienced in Canada. Testing time would not be appreciably lengthened as these conditions readily occur during outdoor testing. The requirement for fixed slope may be harder to achieve during outdoor testing but may be suitable if indoor testing with a solar simulator is utilized.

During outdoor testing, steady state conditions were characterized by solar radiation intensities greater than 900 W/m². Limiting the acceptable range of solar radiation to values above 900 W/m² should have little or no effect on testing time but will also reduce the scatter in test data slightly.
10. CONCLUSIONS

1. Theoretical analysis has been performed for a single glazed flat-plate solar collector. Results indicate that thermal efficiency is highly dependent on meteorological and operational conditions.

2. The simple Hottel, Whillier, Bliss model of solar collector performance does not account for these varying conditions.

3. A computer program has been developed that may be used by system designers to evaluate the effect of meteorological conditions. This modified 1-dimensional analysis adequately represents the performance of flat-plate solar collectors.

4. The relative magnitude of different meteorological effects is a function of solar collector optical and thermal properties.

5. Tests results derived within the ASHRAE test procedures (4), allow for a high degree of variation in the results.

6. Tests conducted over an extended period exhibited less extreme meteorological extremes than those allowed in (4) but this may be location dependent.
7. Tests derived in different climatic locations may differ significantly due to the different meteorological factors present.

8. Solar system designers should take care to ensure that the conditions under which their test data was derived are similar to the planned application or they should be modified to represent the conditions present.

9. Work is required to better determine values of wind induced surface heat transfer coefficient and effective sky temperature.

10. The values of predicted effective sky temperature are extreme. Values for overcast skies should be determined. Clear sky and night time values should be verified. The values predicted may not only be significant to solar collectors but also to building heat loss calculations.

11. If values of $h_w$ and $T_s$ can be accurately determined, normalization techniques may prove practical.

12. The experiments presented attempted to limit systematic errors and sample variability. The results show less deviation in the experimentally derived values of thermal efficiency than those measured at different facilities. The results show less scatter than previous "Round-Robin" test programs.
11. AREAS FOR FUTURE STUDY

The effects of meteorological and operational conditions should be determined for other collector types.

Further research is required on techniques to measure and predict wind induced surface heat transfer coefficient and effective sky temperature. Standard techniques for normalization of test results for a variety of collector types should be developed.

A more accurate representation of collector performance than the Hottel, Whillier, Bliss, model should be developed which accounts for the effects of varying operational and meteorological conditions.
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APPENDIX A

Physical and Optical Specifications of the
Chamberlain Solar Collector
FIGURE
CONSTRUCTION DETAILS FOR CHAIMERLAIN
SOLAR COLLECTOR
MANUFACTURER
CHAMBERLAIN MFG CO.
ELMHURST, ILLINOIS
U.S.A.

GROSS AREA
1.95 m²

APERTURE AREA
1.78 m²

COVER PLATE ASSEMBLY
NUMBER OF GLASS PLATES
1

THICKNESS
3.2 mm

MATERIAL
LOW-IRON, TEMPERED

SOLAR TRANSMITTANCE, τ
0.90

I R EMITTANCE
0.88

ABSORBER PLATE
MATERIAL
MILD STEEL

FLOW CONFIGURATION
19 PARALLEL PASS

COATING
BLACK CHROME

SOLAR ABSORPTANCE, q
0.94

EMITTANCE, ε
0.12

AIR SPACES
BETWEEN COVER AND ABSORBER
10 mm

INSULATION
MATERIAL
GLASS-FIBRE

DENSITY
80 kg/m³

THICKNESS
76 mm

THERMAL CONDUCTIVITY
0.03 W·m⁻¹·°C⁻¹

THICKNESS τ 0.90

TEMPERED GLASS

q 0.94, ε 0.12

19 mm AIR SPACE

STEEL ABSORBER

76 mm GLASS-FIBRE INSULATION

GALVANIZED STEEL BOX

SCHEMATIC OF FLAT-PLATE COLLECTOR
APPENDIX B

COLSM Simulation

Computer Program Listings
C PROGRAM MCH1M
C
DIMENSION F(500),Y(500),CNDM(10)

IMPLICIT HEAT(A-H,J-K)

IMPLICIT LOGICAL(A-J)

DATA (E(1),T(1),L(1),W(1),F(1))=1.0,0.0,1.0,1.0,1.0

DATA CNDM/45.0,40.0,10.0,10.0,10.0,10.0,10.0,10.0,10.0,10.0/

FIX=1.415925

H=1.79

IMP=1.0

TP=1.0

C

10 CALL CLEAR

20 CALL CLEAR

30 CALL CLEAR

40 CALL CLEAR

50 CALL CLEAR

60 CALL CLEAR

70 CALL CLEAR

80 CALL CLEAR

90 CALL CLEAR

C

10 CONTINUE

20 FORMAT(* "DO YOU WISH TO CHANGE TEST CONDITIONS? (Y/N):" ' )

30 ACCEPT Y,N

40 FORMAT(A1)

50 IF(LAND,NE.2.0) GO TO 20

60 WRITE(7,90) IFANS

70 WRITE(7,91) IFANS

80 WRITE(7,92) IFANS

90 DO TO 90

100 CONTINUE

C

100 WRITE(7,100)COND(1)

200 WRITE(7,200)COND(2)

300 WRITE(7,300)COND(3)

400 WRITE(7,400)COND(4)

500 WRITE(7,500)COND(5)

600 WRITE(7,600)COND(6)

700 WRITE(7,700)COND(7)

800 WRITE(7,800)COND(8)

900 WRITE(7,900)COND(9)

C

100 CONTINUE

C

200 CONTINUE

300 CONTINUE

400 CONTINUE

500 CONTINUE

600 CONTINUE

700 CONTINUE

C

800 CONTINUE

900 CONTINUE

C

100 WRITE(7,100)COND(1)

200 WRITE(7,200)COND(2)

300 WRITE(7,300)COND(3)

400 WRITE(7,400)COND(4)

500 WRITE(7,500)COND(5)

600 WRITE(7,600)COND(6)

700 WRITE(7,700)COND(7)

800 WRITE(7,800)COND(8)

900 WRITE(7,900)COND(9)
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0001  SUBROUTINE COLSM (TYP, TILT, ANG1, 6, FRACD, M1, R,
TAC, C1,2, H, AF, 0, H1, G, PMITA, TS)
C PROGRAM COLSM  VERSION 3.0  LAST UPDAT SEPT 9.1982
C SUBROUTINE COLSM  SUBROUTINE COLSM  SUBROUTINE COLSM
C THIS PROGRAM CALCULATES THE INTERNAL PERFORMANCE OF THE CHAMBER &
C SOLAR COLLECTOR AT DIFFERENT OPERATIONAL AND ATMOSPHERIC CONDITIONS.
C Specify Simulation Parameters
C
C IMPLICIT REAL (A-H, I-J, Z)
C REAL (L, M, I, J, K, N)
C PARAMETER (ATMOP=5.47OF-B
C PI=3.14159265
C T0=0.05
C
C Specify Collector Characteristics
C
C Dimensions (in MKS units)
C
C A0=1.79
C AP=1.79
C A1=1.79
C A2=1.93
C UH=0.002537
C UI=0.0762
C IPC=0.019
C D=0.0407
C I=0.08
C W=0.0432
C
C Optical Properties
C
C N=1.518
C KL=0.0035
C TRL=0.02
C
C Heat Transfer Properties
C
C Cover
C EM=0.88
C Absorber Plate
C CSAT=0.000927

0022  C NT=19.0
0023  C WPER=0.085
0024  C FMP=0.12
0025  C ALFAP=0.95
0026  C F=0.08
C Collector Housing
0027  C KI=0.03
0028  C UE=0.1
C Environmental and Operating Conditions
C Environmental Conditions (Temperature in Degree K)
0029  C TTT=273.0+CEI
0030  C TA=273.0+TAC
0031  C PATM=1.0
C Operational Conditions
0032  C SNFR=MFLR/AP
0033  C 80 CONTINUE
C Heat Transfer Model and Analysis
C
C 80 CONTINUE
0034  C TPH=FII+2.0
0035  C CALL TOLP (TOL, EM, FMP, FMP, ALFAP, TRLW, TILT, TPG,
> TAP, WSNF, UPLI, IPC, TAC, TS)
0036  C UTOP=(AP/AI)/UTOP.
C Determine the Value of the Overall Heat Loss Coefficient
C Back Heat Loss Coefficient
C Overall Heat Loss Coefficient
C
C =U=I+B+H+H+H
C
C Calculate Optical Properties
C
C CALL STAFF (M, ALFAP, ANGLE, TILT, FRACD,
> TAC, FII, TPH, TAC, TAC, TAC, C1,2, H1, G, PMITA, TS)
C
C 250 CONTINUE
C **************************** CALCULATION OF USIFUL ENERGY COLLECTED *****************************
C
C 0
C 044  QU=AP4RF*(Q1+APFR)*FU+FI*(TFI-TA))
C 045  IF(QU<4.0) GO TO 500
C
C 046  C 047  TFM=FIX((Q1/QU)/(UL+FU))#1.-(FR/FPRHE))
C 048  ERR=ABS(TFM-TF)
C 049  IF(ERR<0.001) GO TO 350
C 050  TF=TFM
C
C 051  C 052  C 053  IF(FRR<0.001) GO TO 400
C 054  TFM=TF
C 055  GO TO 90
C
C 056  400  CONTINUE
C
C 057  401  C 058  C 059  NITA=(Q1/(QU+FU))
C 060  PHTA=NITA*100.
C
C 061  C 062  C 063  WRITE(6,240)TFI,TFF,UF,UF+PR,HF,UPC,UCR,UTOP,UL,FPRHE,
C 064  TFR,TAE FF,AD T1,PHTA
C
C 065  240  FORMAT(2X,1F6.1,2F1.1,F8.1,F6.1,F6.1,F6.1,F7.3,3X,4F6.2,3X,2F6.3,
C 066
C
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C
C 067  2X,2F6.3,F6.1)
C
C 068  WRITE(6,400)TFM,CP+BC,H,H,F,AD1SC
C
C 069  400  FORMAT(1X,F7.3,10X,F9.4,3X,F9.3,3X,F9.5,3X,F10.8)
C
C 070  WRITE(6,461)VAVE,RE,HUF,HF,UD
C
C 071  461  FORMAT(2X,VAVE,R,E,HUF,HF,RE,HF,UD)
C
C 072  RETURN
C
C 073  END
SUBROUTINE TOP1 (TPH, EMP, ENG, PC, ALFA, TPM, TAnalysis, UPA, UCA, TIR)
C THIS ROUTINE CALCULATES THE VALUE OF THE TOP LOSS COEFFICIENT
C
C INPUT:
C MEAN ABSORBER TEMPERATURE AND CALCULATE HEAT TRANSFER RATES.
C IT THEN ITERATES TO REFINE ESTIMATES TO WITHIN SET TOLERANCES.
C
C IMPLICIT REAL (A-H, I-J, Z)
C NEI INITIAL VALUES
C INI = 1.0
C S = 0.025E-8
C TD = 9.0 TC = (TPH+TA)/2.0
C CALL TSAT(TA, EMP, ENG, TIL, R, T5)
C HEAT TRANSFER COEFFICIENT (COVER I SURROUNDING ENVIRONMENT)
C HM = 5.76 (J/SEC/WM)
C HM = 5.253 (J/SEC/WM)
C INCA = (TC-FA) DCA = 0.0
C HRCA = FMC*SIGMA*(TPH**4 - TSO**4)/(DCA)
C IF (HRCA < 0.0) HRCA = 0.0
C HRCA = HM*HRCA
C HEAT TRANSFER COEFFICIENT (ABSORBER PLATE TO COVER)
C CALL CNHPC(TPM, TPM, TPM, PC, ALFA, TIL, UPA)
C OVERALL HEAT TRANSFER COEFFICIENT (ABSORBER PLATE TO COVER)
C UPA = UPHC/ALPHA
C OVERALL TOP LOSS HEAT TRANSFER COEFFICIENT (PLATE TO ATMOSPHERE)
C UTOP = 1.0/(1.0/UPA + (1.0/UCA)) + HUPB
C
C *** CHECK IF ASSUMED COVER TEMPERATURE IS WITHIN TOLERANCES ***
C IF NOT, SET ASSUMED COVER TEMPERATURE TO THE CALCULATED VALUE AND

C ITERATE AGAIN UNTIL VALUE IS WITHIN TOLERANCES.
C
C TMC = TMC - (UTOP + (TPM - FA))/UPA
C RPM = ABS(TMC - TC)
C IN = IN
C IF (TPM < 100) GO TO 200
C TC = TMC
C GO TO 100
C RETURN
C FIN

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SUBROUTINE SKY(TA,KW1,TII,NS,TS)
C SUBROUTINE TO CALCULATE THE EFFECTIVE SKY TEMPERATURE OF THE
C ENVIRONMENT AS SEEN BY A SURFACE INCLINED AT AN ANGLE "TILT"
C TO THE HORIZONTAL.
C
TA   =AMBIENT AIR TEMPERATURE (DEG. K)
KII  =TOTAL EMITTANCE OF THE GROUND TO LONGWAVE RADIATION
TILT =TILT ANGLE OF SURFACE TO HORIZONTAL (DEG.)
SLOPE =TILT ANGLE IN RADIANS
THETA =REFRACTION ANGLE (RADIANS)
RA   =ATMOSPHERIC RADIATION INCIDENT ON HORZ. SURFACE
RAR  =ATMOSPHERIC RADIATION INCIDENT ON INCLINED SURFACE
RS   =TOTAL LONGWAVE RADIATION INCIDENT ON SURFACE
K1   =CONSTANT FOR RA CALC. FROM COLE
K3   =CONSTANT FOR RAR CALC. FROM COLE
SIGMA =STEPHAN-BOLTZMANN CONSTANT
TS   =EFFECTIVE SKY TEMPERATURE (DEG.K)
TG   =TEMPERATURE OF GROUND, ASSUMED = TA

0002  IMPLICIT REAL(A-H,J-L,Z)
0003  PI=3.14159265
0004  SIGMA=5.6700*10**-8

C CALCULATE SLOPE ANGULAR (RADIANS)
0005  SLOPE=TII/PI/180.0
C CALCULATE RA
0006  RA=1.094*0.03*SIGMA*(TA**4)
C CALCULATE RAR
0007  X=TILT
0008  K1=1.00000-4.46622F-06X-7.47489E-05X**2+1.42192F-04X**3
0009  >1.40696E-09X**4
0010  K3=9.003035E-05X+1.0221E-04X**2+1.71858E-06X**3
0011  >7.17531E-08X**4
0012  RA=RASK1+0.09K3*SIGMA*(TA**4)
C CALCULATE RBS
0013  TG=TA
0014  RBS=ENG*SIGMA*(TA**4)*((1.0-COS(SLOPE))/2.0)

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C CALCULATE RS
0015  RS=RBS+RBS
C CALCULATE TS
0016  TS=(RS/SIGMA)**0.25
C
WRITE(6,100)TA,KW1,K3,RS,RBG,KW1
100  FORMAT(8F10.3)
0017  RETURN.
0018  END
SUBROUTINE CH,P(m,IC:T,M,PC:AL,TAP,TIL,HPC)
C THIS ROUTINE CALCULATES THE VALUE OF THE CONVECTIVE HEAT 1/2
C COEFFICIENT BETWEEN THE ABSORBER PLATE AND COVER.
C
C IMPLICIT REAL (A-H,I-Z)
C DATA INITIAL VALUES
C PT=1.141592654
C SIGMA=5.6703-6
C SLOPE=1.0,1.0
C CONVECTIVE HEAT TRANSFERT COEFFICIENT (ABSORBER PLATE TO COVER)
C..............................................
C AIR LAYER PROPERTIES (ASSUMED AT A TEMPERATURE OF TALM)
C........................................................................
C TALM=(TPM+TC)/2.
C DTAL=TPM-TC
C XC=PC
C XR=100.,/TALM
C CALCULATE RADIATION NUMBER
C RAR=7737.6*((1.47*XA)**2)*(XAR**4)*TALM*((100.*XC)**2)*(PAR**2)
C CALCULATE THERMAL CONDUCTIVITY OF AIR AT THE ASSUMED TEMPERATURE
C KR=0.00025B*(TALM**1.5)/(TALM+290.)
C CALCULATE THE NUSSEIT NUMBER (BY TEMS)
X1=(1.)+(708.,/(RASCONS(SLOPE))
X1=(1.,1.0.)/X1=0.0
X1=(1.)+(51.,1.0.)/X1=0.0
X1=(1.,1.0.)/X1=0.0
C WRITE(6,20) X1,X1
C 70 FORMAT(3F10.5)
C HPC=MUS(NAR/PC)
C ..............................................
C WRITE(6,80)TPM+TC,TALM,PC,TIL,HPC
C WRITE(6,40)TPM+TC,TIL,PC
C 80 FORMAT(11E11.4)
C 200 RETURN
FEND
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0001  SUBROUTINE STAFF(AM,ALAP,AL,ALP,TILT,FRACD,RF,UP,T,TEAFF)
C SUBROUTINE TO CALCULATE THE OPTICAL PROPERTIES OF THE GLAZING
C
C M - INDEX OF REFRACTION FOR GLAZING
C N - PRODUCT OF EXTINCTION COEF. X COVER THICKNESS
C ANGIF - ANGLE BETWEEN GLAZING SURFACE NORMAL TO KERAM RAY, (DEG.)
C TILT - SOLAR COLLECTOR TILT IN DEGREES
C FRACD - FRACITION OF DIFFUSE RADIATION ON THE COLLECTOR TILT
C RF - RADIANCE INCIDENT TO SOLAR RADIATION
C RFCD - RADIANCE INCIDENT TO AMB. PLATE TO ATMOSPHERE
C T - HEAT LOSS COEFF. TO AMB. PLATE TO COVER
C TRADC - AVERAGE TRANSMITTANCE OF COVER
C TAEFF - EFFECTIVE TRANSMITTANCE ABSORPTION PRODUCT
C TRGS - TRANSMITTANCE OF GLAZING DUE TO ABSORPTION ONLY
C AREF - EFFECTIVE INCIDENCE ANGLE FOR AMB. PLATE SKY DIFFUSE
C AREFD - EFFECTIVE INCIDENCE ANGLE FOR AMB. PLATE SKY DIFFUSE
C TEAFF = EFFECTIVE TRANSMITTANCE TO GROUND AND SKY DIFFUSE
C
C IMPLICIT REAL(A-H,O-Z)
C P1=3.14159265
C SLOPE=0.502515/180.0
C HSP=1.0-(COS(SIPI)/P1)^2.
C SSF=(1.0+DOSI)/2.
C
C TRANSMISSION OF GLAZING TO BEAM RADIATION
C
0007  CALL OPTICAL(N,M,ANGLE,TRANSC,RFCD,TRGS)
0008  AREFD=(3.0,TILT)
C WRITE((A,20),N,M,ANGLE,TRANSC,RFCD,TRHS,ARF)
C 20  FORMAT(7F10.3)
C C CALCULATE TRANS. OF GLAZING TO SKY AND GROUND DIFFUSE RADIATION
C
0009  CALL OPTICAL(N,M,ANGLE,TRANSC,RFCD,TRHS,ARF)
C C SKY SOURCE DIFFUSE
C C EFFECTIVE INCIDENCE ANGLE FOR SKY DIFFUSE
C 0009  AREFD=0.45,0.1385TILT-0.014978(TILT)**2.
C C CALCULATE TRANSMITTANCE TO SKY DIFFUSE
C 0010  CALL OPTICAL(N,M,ANGLE,TRANSC,RFCD,TRHS,ARF)

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C GROUND SOURCE REFLECTED DIFFUSE
C EFFECTIVE INCIDENCE ANGLE FOR GROUND DIFFUSE
0011  AREFD=0.45,0.578+TILT-0.002+TILT**2
C CUBE TRANSMITTANCE TO GROUND SOURCE DIFFUSE
C
0012  CALL OPTICAL(N,M,ANGLE,TRANSC,RFCD,TRHS,ARF)
C WRITE((A,30),N,M,ANGLE,TRANSC,RFCD,TRHS,ARF)
C 30  FORMAT(7F10.3)
C C EFFECTIVE TRANSMITTANCE TO GROUND AND SKY DIFFUSE
C
0013  TEAFF=(TRHS+ARFCD+RFCD)/RF
C C EFFECTIVE DIFFUSE TRANS. OF GLAZING DUE TO ABSORPTION ONLY
C 0014  TRAEFF=+TRAEFF+(TFCD+TRHS+ARFCD)
C 0015  AREFF=0.45,1.0-ARF
C WRITE((A,35),TAEFF,ARF)
C 35  FORMAT(7F10.3)
C C TRANSMISSION X ABSORPTION PRODUCT
C
0016  NFAB=0.0
C 0017  CALL OPTICAL(N,M,ANGLE,TRANSC,RFCD,HDIF)
C TRNADD=TEAFF/(1.0-NFAB)
C TRNADD=(TEAFF/(1.0-NFAB))/RF
C C EFFECTIVE TRANSMITTANCE X ABSORPTION PRODUCT
C
0020  TAEFF=TRNADD*ARF
C 0021  TRAEFF=TRNADD*ARFCD
C 0022  TAEFF=(1.0-ARFCD)*TAEFF
C WRITE((A,25),TAEFF,TRAEFF)
C 25  FORMAT(7F10.3)
C C EFFECTIVE TRANSMITTANCE X ABSORPTION PRODUCT
C 0023  RETURN
C 0024  END
SUBROUTINE UPTCN(MXIC, ANGLE, TRANCE, REFC, KL, KG)
C SUBROUTINE TO CALCULATE THE OPTICAL PROPERTIES OF THE GLAZING
C
N = INDEX OF REFRACTION FOR GLAZING
C KLC = PRODUCT OF EXTINCTION COEF. & COVER THICKNESS
C ANGLE = (ANGLE) BETWEEN GLAZING SURFACE MKMRAI & MFAIR RAD. (DEG.)
C IH = INCIDENT ANGLE (ANGLE) XPRNED IN RADIANS
C THETA = REFRACTION ANGLE (RADIANS)
C KUMP = AIR/GLASS INTERFACE REFLECTIVITY-PARALLEL COMP.
C ROWPAR = AIR/GLASS INTERFACE REFLECTIVITY-PERCPENDICULAR COMP.
C K = TRANSMITTANCE OF GLAZING DUE TO ABSORPTION ONLY
C REFF = REFLECTANCE OF COVER FOR BOTH COMP. OF POLARIZATION
C REFP = REFLECTANCE FOR GLAZING PERPENDICULAR COMP.
C TRANF = TRANS. OF GLAZING DUE TO ABSORP. AND REFLECTION
C TRANC = AVERAGE TRANSMITTANCE OF COVER
C
IMPLICIT REAL(A-H, I-J, K)
PI = 3.14159265

C CALCULATE INCIDENT ANGLE (RADIANS)
IF(ANGLE.GT.90.) THETA = PI/2.
THETA = ANGLE*PI/180.0
IF(THETA) 30, 40, 10

C CALCULATE REFRACTION ANGLE (SNEHL'S LAW)
10 THEA = SIN(THETA)/N
TEMP = SQRT(1.0 - THEA**2)
THETA = ARC(TMP/THETA2)/TEMP

C CALCULATE REFLECTIVITY AT AIR/GLAZING INTERFACE FOR PARALLEL
C AND PERPENDICULAR COMPENTIONS (FRESNEL'S FORMULA)
RPE = SIN(THETA2)/SIN(THETA1+THETA2)
NPE = RPE*COB(THETA1+THETA2)/COB(THETA1-THETA2)

C FOR THE CASE THETA1 = 0, THEN THETA2 = 0 AND ROWPAR = ROWPER GIVEN BY:
40 ROWPAR = N(-1.)/(N+1.)*R2
50 CONTINUE

C CALCULATE TRANSMITTANCE OF GLAZING DUE TO ABSORPTION OF GLAZING
TRANC = EXP(-1.8KLC/COB(THETA2))
SIRC = TRANC
C REFLECTANCE VALUES (RDL*W OF STOKES EONS FOR MULTIPLE REFLECTIONS)
C FOR THE COVER
REFF = ROWPAR*REFF(+1.8TRANG(1.0-ROWPAR)*SIRC)/(1.0-(ROWPAR)*SIRC)
REFP = ROWPAR*REFP(+1.8TRANG(1.0-ROWPER)*SIRC)/(1.0-(ROWPER)*SIRC)
C AVERAGE VALUES OF REFLECTANCE FOR THE COVER
C TRANSN = (TRANC*TRANG+TRAP)*SIRC
SIRC = RETURN
C
SUBROUTINE APFR( TYP, SMFR, SFR, SF, HF, NUF, CP, CPFRM, FRP )
C THIS SUBROUTINE CALCULATES THE VALUE OF FR FOR THE CHAMBER/PLATE
C COLLECTOR. THE RELATIONSHIP FOR UN SHOULD BE CHANGED TO SUIT.
C OTHER COLLECTION TYPES IF THEY ARE TO BE EVALUATED.
C
C TYP -FLUID TYPF., [FL; 1-L50/50; H2O; H2O; H20]
C FM -MEAN FLUID TEMPERATURE, K
C SMFR -SPECIFIC MASS FLOW RATE, kg/m
C SF -SPECIFIC FLUID HEAT DENITY, kg/m
C HF -COLLECTOR HEAT LOSS COEFF., W/m
C FR -ABSORBENT PLATE AREA, m
C CSAT -COLLECTOR CROSS-FICTIONAL AREA OF FLOW TUBES, m
C NF -NUMBER OF FLOW TUBES
C DH -HYDRAULIC DIAMETER OF FLOW TUBES, m
C I -ABSORBENT PLATE EFFECTIVE LENGTH, m
C W -WIDTH OF FIN AND TUBE, m
C NPER -NUMBER OF PERIMETER LENGTH OF FLOW TUBES, m
C NF -HEAT TRANSFER COEFF., FLUID TO TUBE WALL, W/m
C TAY -THERMAL NUMBER OF FLUID
C NUF -NUSELT NUMBER FOR FLUID
C CP -SPECIFIC HEAT OF FLUID, J/kg
C CPFRM -COLLECTOR EFFICIENCY FACTOR
C HF -HEAT REMOVAL FACTOR FOR COLLECTOR
C FR -FIN EFICIENCY OF ABSORBANT PLATE
C FR -PLATE DIMENSION
C
IMPLICIT REAL(A-H,J-Z)
C
SUBROUTINE TO CALCULATE THE HEAT REMOVAL FACTOR.
C
** UPRM** COLLECTOR EFFICIENCY FACTOR, F' ** UPRM**
C
CALCULATE HEAT TRANSFER FLUID PROPERTIES
CALL FPFRM(TYP, TFM, CP, CPFRM, DENS, ABVISC, KF, KVISCE)

VAR = (SF/SMFR)/(DENS*CSAT*M)
WRITE (4, 10D9)
FORMAT (F4,2)
CALCULATE THE REYNOLDS NUMBER FOR THE HEAT TRANSFER FLUID
RE = (DENS*VAR*SMFR)/ABVISC

SUBROUTINE H: PROP(TYP,TM,CF,PR,DFNS,AVISC,KF,KVISC)
C THIS SUBROUTINE CALLS ATHER THE HEAT TRANSFER FLUID
C PROPERTIES BASED ON FLUID TEMPERATURE
C
C *** FLOW ***
C TM - MEAN FLUID TEMP IN K
C TMFC - MEAN FLUID TEMP IN DEG. C
C CP - FLUID SPECIFIC HEAT, J/Kg C
C CF - FLUID THERMAL CONDUCTIVITY, W/m C
C KVISC - FLUID KINEMATIC VISCOSITY, cm^2/sec
C AVISC - FLUID ABSOLUTE VISCOSITY, kg/m sec
C PR - FLUID PRANDTL NUMBER
C
C IMPLICIT REAL(A-H,I-Z)
0003 TFC = TM - 273.0
0004 IF (TYP.EQ.1.) GO TO 100
C
C THE FOLLOWING EQUATIONS ARE FOR THE PROPERTIES OF WATER
C BASED ON THE FLUID TEMPERATURE IN DEGREES CELSIUS
0006 CF = .0149589#TFMC#2 + 1.37698#TFMC# + 4210.02
0007 CF = -.3453201#TFMC#2 + 1.054377#TFMC# + 1.04594
0008 CF = -.1505481#TFMC#2 - .0670344#TFMC# + 1.00041
0009 CF = -.419790#TFMC#2 + 1.01792#TFMC# - 2.01957#TFMC# - 2.419790#TFMC#2
0010 CF = 9.13364#TFMC# - 1.34136#TFMC#2 + 1.7415
0011 CF = VISC = TFMC#0.7
0012 CF = KF = CF#AVISC/KF
0013 CF = GO TO 101
C
C THE FOLLOWING EQUATIONS ARE FOR THE PROPERTIES OF ICE AT 50/50 MIXTURE WITH WATER BY VOLUME BASED ON THE FLUID
C TEMPERATURE IN DEGREES CELSIUS
0014 CF = -.0137147#TFMC#2 + 5.9718#TFMC# + 3148.08
0015 CF = -.100505#TFMC#2 - 6.37510#TFMC# + 1.4369
0016 CF = -.0018512#TFMC#2 - 4.65208#TFMC# + 10.7774
0017 CF = -.00020124#TFMC#4 + 0.2044#TFMC#2 - 8.52159#TFMC# - 3.28159#TFMC#2
C 100 RETURN
C 101 FND
C
SUBROUTINE PSETUP(FX, FY, IPOINTS)

INTEGER FX(500), FY(500)
LOGICAL ICI(1)
DATA IETM/'H'/, IFTS/'S'/, IFIG/'Y'/

CALL CLEAR

IF(IETM) FORMAT(* + HARD COPY (H) OR SCREEN (S) PLOT? Y, N *)
ACCEPT 300, IANS
IF(IANS.NE.151) GO TO 10

ISCREEN = .FALSE.
IF(IANS.NE.151) GO TO 10
ISCREEN = .TRUE.

XPAD = 10.0
YPAD = 7.0
XAXIS = 5.0
YAXIS = 5.0
XORD = 0.75
YORD = 0.0

GO TO 40

IF(ANS.NE.151) GO TO 10

CONTINUE

CALL CLEAR

50 FORMAT(* + DO YOU WISH TO PLOT THE AXIS Y, M? Y, N *)
ACCEPT 300, IANS
IF(IANS.NE.1FTY) GO TO 4000

CONTINUE

XSTP = 0.02
YSTP = 20.0

END
APPENDIX C

Energy Balance for Chamberlain Solar Collector

Run of COLSM with:

\[ \dot{h}_w = 5 \text{ and } 30 \text{ W/m}^2\text{°C} \]
\[ U_e = 0.1 \text{ W/m}^2\text{°C} \]

where:

\[ q_e = F U A (T_{e} - T_{a}) \]

Output variable description

**EFF** - Collector Thermal Efficiency

**DT/I** - \( \frac{(T_{fi} - T_{a})}{I} \)

**QA** - Rate of Energy Absorbed by Absorber Plate

**QU** - Rate of Energy Collection by Collector

**QL** - Rate of Thermal Energy Loss From Collector Case

**PTL** - Percent of Total QL Due To Top Losses

**PBL** - Percent of Total QL Due to Back Losses

**PEL** - Percent of Total QL Due to Edge Losses
### OPERATIONAL CONDITIONS FOR SIMULATION

- **TILT**: INCID. ANGLE SOLAR INSOL.
- **DIFF.FRAC.**: FLOW RATE
- **AIR TEMP.**: FLUID TEMP.
- **W**: 100.0 W/sq.m

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<tr>
<td>SOLAR INSOL.</td>
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<tr>
<td>DIFF.FRAC.</td>
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<td>FLOW RATE</td>
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<td>AIR TEMP.</td>
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<tr>
<td>FLUID TEMP.</td>
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**GROUND EMITTANCE 0.9, GROUND REFLECTIVITY 0.2**

### DETAILED HEAT TRANSFER ANALYSIS

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**EFFECTIVE RAY TEMP. DURING TEST WARM**: 54.7 deg.C

### OPERATIONAL CONDITIONS FOR SIMULATION

- **TILT**: INCID. ANGLE SOLAR INSOL.
- **DIFF.FRAC.**: FLOW RATE
- **AIR TEMP.**: FLUID TEMP.
- **W**: 100.0 W/sq.m

**GROUND EMITTANCE 0.9, GROUND REFLECTIVITY 0.2**

### DETAILED HEAT TRANSFER ANALYSIS

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**EFFECTIVE RAY TEMP. DURING TEST WARM**: 45.3 deg.C
APPENDIX D

Data for Runs of COLSM
To Determine the Effect
of Meteorological Factors on
Thermal Performance
### COLTIM PERFORMANCE SUMMARY

**OPERATIONAL CONDITIONS FOR SIMULATION**

- **TILT**: 45.0 degrees
- **INCIDENCE ANGLE**: 0.0 degrees
- **SOLAR INTENSITY**: 1000.0 W/m²
- **DIFFUSION**: 0.20
- **FRAC.**
- **FLOW RATE**: 0.036 kg/sec
- **AIR TEMP.**
- **FLUID TEMP.**
- **WIND**

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**GROUND EMITTANCE 0.9; GROUND REFLECTIVITY 0.2**

**DETAILED HEAT TRANSFER ANALYSIS**

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**EFFECTIVE SKY TEMP. DURING TEST WAS -7.4 degrees C**
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#### Effective Sky Temp. During Test was: -11.2 deg.C
### Detailed Heat Transfer Analysis

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**Effective Sky Temp. During Test Was:** 3.4 deg.C
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**Effective Sky Temp. During Test was:** 7.4 deg.C

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**EFFECTIVE SKY TEMP. DURING TEST:** -7.4 deg C

**OPERATIONAL CONDITIONS FOR SIMULATION**

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**EFFECTIVE SKY TEMP. DURING TEST:** -7.4 deg C
### DETAILED HEAT TRANSFER ANALYSIS

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|---|---|--------|--------|------|-----|------|-------|------|--------|------|------|-------|-------|------|------|------|------|
| 286.0 | 269.9 | 27.2 | 3.8 | 51.5 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 293.0 | 304.5 | 29.2 | 3.8 | 27.5 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 298.0 | 309.1 | 33.2 | 3.8 | 24.4 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 303.0 | 313.7 | 37.3 | 3.8 | 21.9 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 308.0 | 318.3 | 41.8 | 3.8 | 19.7 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 311.0 | 322.8 | 46.5 | 3.8 | 17.9 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 316.0 | 327.4 | 51.7 | 3.8 | 16.2 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 321.0 | 332.0 | 57.4 | 3.8 | 14.7 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 326.0 | 336.6 | 63.6 | 3.8 | 13.4 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 330.0 | 341.1 | 70.4 | 3.8 | 12.2 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 338.0 | 345.7 | 77.9 | 3.8 | 11.1 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 340.0 | 350.3 | 84.1 | 3.8 | 10.1 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 348.0 | 354.9 | 95.1 | 3.8 | 9.2 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 353.0 | 359.4 | 104.9 | 3.8 | 8.4 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 358.0 | 364.0 | 113.5 | 3.8 | 7.7 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 360.0 | 368.6 | 124.7 | 3.8 | 7.1 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 368.0 | 373.2 | 138.4 | 3.8 | 6.5 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 373.0 | 377.7 | 150.2 | 3.8 | 6.1 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 378.0 | 382.3 | 161.7 | 3.8 | 5.7 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
| 381.0 | 384.9 | 177.5 | 3.8 | 5.4 | 354.7 | 3.42 | 82.87 | 3.22 | 3.75 | 0.985 | 0.958 | 0.815 | 0.016 | 78.1 |
### OPERATIONAL CONDITIONS FOR SIMULATION

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| GROUND EMITTANCE 0.9 | GROUND REFLECTIVITY 0.2 |

#### DETAILED HEAT TRANSFER ANALYSIS

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#### EFFECTIVE SKY TEMP. DURING TEST WAS: 7.4 deg.C

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| GROUND EMITTANCE 0.9 | GROUND REFLECTIVITY 0.2 |

#### DETAILED HEAT TRANSFER ANALYSIS

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#### EFFECTIVE SKY TEMP. DURING TEST WAS: 7.4 deg.C
### Colsin Performance Summary

#### Ground Emissivity and Reflectivity

**Ground Emissivity 0.9, Ground Reflectivity 0.2**

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| 293.0 304.6 29.2 3.8 27.5 |
| 298.0 309.2 33.2 3.8 24.4 |
| 303.0 317.9 37.4 3.8 21.9 |
| 308.0 318.2 41.4 3.8 18.7 |
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| 318.0 327.7 51.9 3.8 16.1 |
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| 363.0 369.3 127.6 3.8 7.0 |
| 368.0 374.0 139.4 3.8 6.3 |
| 373.0 378.6 151.2 3.8 6.0 |
| 378.0 383.2 162.7 3.8 5.3 |
| 383.0 387.9 173.3 3.8 5.3 |

**Effective Sky Temp. During Test: -7.4 deg.C**

### Operational Conditions for Simulation

**Diff. Frac., Flow Rate, Air Temp., Fluid Temp., HW**

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| 293.0 303.5 29.2 3.8 27.5 |
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| 378.0 383.2 162.7 3.8 5.3 |
| 383.0 387.9 173.3 3.8 5.3 |

**Effective Sky Temp. During Test: -7.4 deg.C**
**COSLIN PERFORMANCE SUMMARY**

**OPERATIONAL CONDITIONS FOR SIMULATION**

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**GROUND EMITTANCE 0.9**  GROUND REFLECTIVITY 0.2

**DETAILED HEAT TRANSFER ANALYSIS**

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**EFFECTIVE SKY TEMP. DURING TEST WAS:** -7.4 deg.C

**OPERATIONAL CONDITIONS FOR SIMULATION**

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**GROUND EMITTANCE 0.9**  GROUND REFLECTIVITY 0.2

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**EFFECTIVE SKY TEMP. DURING TEST WAS:** -7.4 deg.C

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**GROUND EMITTANCE 0.9**  GROUND REFLECTIVITY 0.2

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**EFFECTIVE SKY TEMP. DURING TEST WAS:** -7.4 deg.C
COSLH PERFORMANCE SUMMARY

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GROUND EMITTANCE 0.94  GROUND REFLECTIVITY 0.2

DETAILED HEAT TRANSFER ANALYSIS

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GROUND EMITTANCE 0.94  GROUND REFLECTIVITY 0.2

DETAILED HEAT TRANSFER ANALYSIS

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EFFECPIVE SKY TFMP. DURING TEST WAS: -7.4 deg.C
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**Effective Sky Temp. During Test War:** -7.4 deg.C

**Operational Conditions for Simulation:**
- **Tilt Incidence Solar Insol.:** Diff. frac. flow rate air temp. fluid temp. HW
- **45.0 deg. 40.0 deg. 720.0 W/sq.m. 0.33 0.036 K/sec 10.0 deg.C 10.0 deg.C 15.0 W/cm² K**

### Ground Emittance 0.9

**Ground Reflectivity 0.2**

---

**Operational Conditions for Simulation:**
- **Tilt Incidence Solar Insol.:** Diff. frac. flow rate air temp. fluid temp. HW
- **45.0 deg. 75.0 deg. 490.0 W/sq.m. 0.50 0.036 K/sec 10.0 deg.C 10.0 deg.C 15.0 W/cm² K**

**Ground Emittance 0.9**

**Ground Reflectivity 0.2**
### COULSM PERFORMANCE SUMMARY

**OPERATIONAL CONDITIONS FOR SIMULATION**

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**GROUND EMITTANCE 0.9, GROUND REFLECTIVITY 0.2**

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| EFFECTIVE SKY TEMP. DURING TEST | 1.0 | -44.5 deg.C |

**OPERATIONAL CONDITIONS FOR SIMULATION**

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**GROUND EMITTANCE 0.9, GROUND REFLECTIVITY 0.2**

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| EFFECTIVE SKY TEMP. DURING TEST | 1.0 | 17.3 deg.C |
### COLSON PERFORMANCE SUMMARY

#### OPERATIONAL CONDITIONS FOR SIMULATION
- TILT: 45.0 deg.
- INCID. ANGLE: 0.0 deg.
- SOLAR INSDL. DIFF.: 1000.0 W/m²
- FRACT.: 0.00 %
- FLOW RATE: 0.50 m³/s
- AIR TEMP.: 10.0 deg.C
- FLUID TEMP.: 10.0 deg.C
- HW: 5.0 W/(m² K)

#### GROUND EMISSANCE 0.9

#### GROUND REFLECTIVITY 0.2

#### DETAILED HEAT TRANSFER ANALYSIS

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#### EFFECTIVE SKY TEMPERATURE DURING TEST:

-7.4 deg.C

#### OPERATIONAL CONDITIONS FOR SIMULATION

- TILT: 45.0 deg.
- INCID. ANGLE: 0.0 deg.
- SOLAR INSDL. DIFF.: 1000.0 W/m²
- FRACT.: 0.00 %
- FLOW RATE: 0.50 m³/s
- AIR TEMP.: 10.0 deg.C
- FLUID TEMP.: 10.0 deg.C
- HW: 5.0 W/(m² K)

#### GROUND EMISSANCE 0.7

#### GROUND REFLECTIVITY 0.7

#### DETAILED HEAT TRANSFER ANALYSIS

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#### EFFECTIVE SKY TEMPERATURE DURING TEST:

-7.4 deg.C
### COLBIN PERFORMANCE SUMMARY

OPERATIONAL CONDITIONS FOR SIMULATION
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- DIFF. FRACT: FLOW RATE
- AIR TEMP.:
- FLUID TEMP.:
- HW: 43.0 deg. 0.0 deg. 1000.0 W/m²

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GROUND EMITTANCE 0.9: GROUND REFLECTIVITY 0.2

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### EFFECTIVE SKY TEMP. DURING TEST WAS:

- 7.4 deg.C
## Coldin Performance Summary

**Operational Conditions for Simulation**

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**Ground Emittance 0.9, Ground Reflectivity 0.2**

## Detailed Heat Transfer Analysis

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**Effective Sky Temp. During Test Was:** -7.4 deg.C
## OPERATIONAL CONDITIONS FOR SIMULATION

**TILT INCID. ANGLE SOLAR INBOL.**

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**GROUNDB EMITTANCE 0.9, GROUND REFLECTIVITY 0.2**

### DETAILED HEAT TRANSFER ANALYSIS

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- **POOR PRINT**
- **Epreuve illisible**

### EFFECTIVE SKY TEMPS. DURING TEST WAR = -7.4 deg.C

**OPERATIONAL CONDITIONS FOR SIMULATION**

**TILT INCID. ANGLE SOLAR INBOL.**

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**GROUNDB EMITTANCE 0.9, GROUND REFLECTIVITY 0.2**

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### OPERATIONAL CONDITIONS FOR SIMULATION

- **TILT**
- **INCID-ANGLE**
- **SOLAR INSOL.**
- **DIFF. FRA.C.**
- **F.L.O.W RATE**
- **A.I.R. TEMP.**
- **F.L.U.I.D TFMP.**
- **W/IN (m/s) K**

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### EFFECTIVE SKY TEMP. DURING TEST WAS: -7.4 deg.C

### OPERATIONAL CONDITIONS FOR SIMULATION

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- **SOLAR INSOL.**
- **DIFF. FRA.C.**
- **F.L.O.W RATE**
- **A.I.R. TEMP.**
- **F.L.U.I.D TFMP.**
- **W/IN (m/s) K**

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**Effective Sky Temp. During Test Was**: -7.4 deg. C
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<td>551.6</td>
<td>6.08</td>
<td>7.89</td>
<td>5.17</td>
<td>3.16</td>
<td>0.987</td>
<td>0.979</td>
</tr>
<tr>
<td>318.0</td>
<td>321.0</td>
<td>175.1</td>
<td>3.8</td>
<td>3.2</td>
<td>556.1</td>
<td>6.38</td>
<td>7.69</td>
<td>5.48</td>
<td>3.19</td>
<td>0.987</td>
<td>0.979</td>
</tr>
<tr>
<td>323.0</td>
<td>325.9</td>
<td>182.9</td>
<td>3.8</td>
<td>2.8</td>
<td>560.6</td>
<td>6.67</td>
<td>7.49</td>
<td>5.80</td>
<td>3.22</td>
<td>0.987</td>
<td>0.979</td>
</tr>
<tr>
<td>328.0</td>
<td>330.8</td>
<td>190.7</td>
<td>3.8</td>
<td>2.3</td>
<td>565.1</td>
<td>6.97</td>
<td>7.29</td>
<td>6.11</td>
<td>3.25</td>
<td>0.987</td>
<td>0.979</td>
</tr>
</tbody>
</table>

**EFFECTIVE SKY TEMP. DURING TEST WAS:** -61.9 deg.C

**OPERATIONAL CONDITIONS FOR SIMULATION**

**TILT INCID. ANGLE SOLAR INCL.**

DIFF.FRAC., FLOW RATE AIR TEMP. FLUID TEMP. HW

75.0 deg. 0.0 deg. 900.0 W/m²

0.10 0.072 kg/sec -35.0 deg.C -35.0 deg.C

5.0 W/(m².K)
## Operational Conditions for Simulation

<table>
<thead>
<tr>
<th>Tilt</th>
<th>Incidence Angle Solar IRRadi</th>
<th>Diff. Fraction</th>
<th>Flow Rate</th>
<th>Air Temp.</th>
<th>Fluid Temp.</th>
<th>HW</th>
</tr>
</thead>
<tbody>
<tr>
<td>45.0</td>
<td>100.0 W/sq.m</td>
<td>0.40</td>
<td>0.036 K/sec</td>
<td>35.0 deg.F</td>
<td>35.0 deg.F</td>
<td>40.0 W/(sq.m)</td>
</tr>
</tbody>
</table>

## Ground Emittance 0.9, Ground Reflectivity 0.2

### Detailed Heat Transfer Analysis

<table>
<thead>
<tr>
<th>Tilt</th>
<th>Fr</th>
<th>Ref</th>
<th>MIF</th>
<th>Fr</th>
<th>HF</th>
<th>UPC</th>
<th>ICA</th>
<th>UTD</th>
<th>UL</th>
<th>FRMEM</th>
<th>FR</th>
<th>TAEFF</th>
<th>DT/1</th>
<th>EFF</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>K</td>
<td>K</td>
<td>Pr</td>
<td>HF</td>
<td>UPC</td>
<td>ICA</td>
<td>UTD</td>
<td>UL</td>
<td>FRMEM</td>
<td>FR</td>
<td>TAEFF</td>
<td>DT/1</td>
<td>EFF</td>
<td></td>
</tr>
</tbody>
</table>

### Effective Sky Temp. During Test Was: 21.4 deg.C

## Operational Conditions for Simulation

<table>
<thead>
<tr>
<th>Tilt</th>
<th>Incidence Angle Solar IRRadi</th>
<th>Diff. Fraction</th>
<th>Flow Rate</th>
<th>Air Temp.</th>
<th>Fluid Temp.</th>
<th>HW</th>
</tr>
</thead>
<tbody>
<tr>
<td>75.0</td>
<td>0.0 0.000 W/sq.m</td>
<td>0.10</td>
<td>0.03A K/sec</td>
<td>-35.0 deg.F</td>
<td>-35.0 deg.F</td>
<td>5.0 W/(sq.m)</td>
</tr>
</tbody>
</table>

## Ground Emittance 0.9, Ground Reflectivity 0.7

### Detailed Heat Transfer Analysis

<table>
<thead>
<tr>
<th>Tilt</th>
<th>Fr</th>
<th>Ref</th>
<th>MIF</th>
<th>Fr</th>
<th>HF</th>
<th>UPC</th>
<th>ICA</th>
<th>UTD</th>
<th>UL</th>
<th>FRMEM</th>
<th>FR</th>
<th>TAEFF</th>
<th>DT/1</th>
<th>EFF</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>K</td>
<td>K</td>
<td>Pr</td>
<td>HF</td>
<td>UPC</td>
<td>ICA</td>
<td>UTD</td>
<td>UL</td>
<td>FRMEM</td>
<td>FR</td>
<td>TAEFF</td>
<td>DT/1</td>
<td>EFF</td>
<td></td>
</tr>
</tbody>
</table>

### Effective Sky Temp. During Test Was: 61.9 deg.C
APPENDIX E

Experimental Equipment

Plate 1 - Photo of NRC/DBR Solar Calorimeter Facility

2 - NRC/DBR Rooftop Test Site

3 - EPPLY Model 8-48 Pyranometer used for determination of incident
total solar radiation on the collector surface

4 - Calibration Certificate for pyranometer used to determine incident
solar radiation

5 - Kaye Data System General Description

6 - Kaye Data System Specifications

7 - Cipher 9-Track Tape Unit, Specifications
   (used for raw data storage)

8 - Shade ring used for determination of diffuse radiation incident on
horizontal surface

9 - Epply Model 50 - pyranomtr used to determine horizontal diffuse
   and total solar radiation
This pyrometer has a radial wirewound-plated (copper-constantan) thermopile enclosed in a glass hemisphere 50 mm in diameter. The thermopile is a differential type with the hot-junction resistors coated with Pernier's black, and the cold-junction resistors coated with barium nitrate. The whiting agent is methylene blue. The spectral response is similar to the 10- and 50-junction models of the 180° pyroheliograph, which this model replaces. The chromed brass case carries a circular spirit level and adjustable leveling screws.

(a) Response:
(b) Linearity of response:
(c) Temperature dependence:
(i) compensated
(-20 to +80°C)
(d) Current response:
(at 10° elevation)
(e) Impedance:
(f) Response time:
63.20
95%
NATIONAL ATMOSPHERIC RADIATION CENTRE

Calibration Certificate No. 76-132

Type of Radiometer: Pyranometer
Manufacturer: EKOLE
Model Number: 8-48
Serial Number: 15321
Calibration Factor, Short-Wave: 7.94 NV/CAL CM² MKS⁻¹
Calibration Factor, Long-Wave: 
Temperature of Calibration: 80°F
Temperature Coefficient: Temperature Compensated
Internal Resistance: 

In Charge of Test: 
Approved: [Signature]

Date of Approval: 3 May 27

NOTES:
1. Short-Wave calibration factors are based on the International Pyrheliometric Scale 1966.
   (Spectral sensitivity of 1913 reduced by 2.0 per cent.)
2. 1 joule cm² = 1 calorie cm⁻² sec⁻¹
   = 0.089 W cm⁻²
   = 221 BTU ft⁻² hr⁻¹
GENERAL DESCRIPTION

The analog "front-end" of the System 6000 is unmatched for reliability, accuracy and flexibility. Greatest emphasis has been placed on reliable operation with ambient conditions and common mode voltage levels typically found in industrial environments. The thermoelectrically heated scanning units will generate a stability of common mode voltages in excess of 400 volts while introducing less than ±1 x 10^-9 of thermal error. A unique electronic reference matches both slope and curvature of the output of a thermocouple to give perfect accuracy of ±0.1°F over a wide range of ambient temperature.

Sensors connected to the System 6000 may be grounded or ungrounded. The use of three-wire, shielded analog circuits and a triple-annealed transformer in the analog power supply provide a high common mode rejection, low noise and a common mode rejection in excess of 140 db.

Analog voltages are converted to digital values by means of a dual-slope integrating DA/DIVA, a high level A/D converter with automatic correction of zero offset. The full-scale value of standard voltage range is ±20000 counts, but the internal full-scale count is ±200000. Dividing the basic count by ten on standard or voltage range eliminates small counting errors.

The greater number of internal counts and a new digital computing circuit, our voltage-to-function converter, provide unusual accuracy and flexibility in conversion of transducer inputs to corresponding values of engineering parameters. For example, thermocouple voltages are converted to temperature units with a conformity of ±0.1°F to the 1971 revised National Bureau of Standards thermocouple tables over the entire normal range of the common thermocouple types. Any system may contain up to three independent converter circuits simultaneously, and the PROM's controlling each circuit are field changeable to allow even greater flexibility.

All system controls are operated by easily accessible front-panel switches; awkward hidden switches and mode gears are not used. The channel identification and data are displayed visually and may be recorded simultaneously on the built-in line printer, or an optional serial recorder.
NUMBER OF CHANNELS:
10 to 990 in 10 channel increments.

MAXIMUM SCAN RATE:
3 readings/sec.

COMMON MODE VOLTAGE:
400 VDC or AC.

STABILITY WITH TIME:
20 mV, 200 mV and 2 V ranges:
30 days
± 1 μV ± 0.005% full scale ± 0.01% reading.
1 year
± 2 μV ± 0.005% full scale ± 0.02% reading.
20 V and 200 V ranges:
30 days
± 2 mV ± 0.005% full scale ± 0.01% reading.
1 year
± 3 mV ± 0.005% full scale ± 0.02% reading.
Thermocouple Types E, J, K and T:
30 days
± 0.1°C ± 0.01% reading.
1 year
± 0.2°C ± 0.02% reading.
Thermocouple Types R and S:
30 days
± 0.4°C ± 0.01% reading.
1 year
± 0.8°C ± 0.02% reading.
20 mV, 200 mV and 2 V ranges:
± 0.1 μV/°C ± 0.001% reading/°C.
20 V and 200 V ranges:
± 0.2 μV/°C ± 0.001% reading/°C.
Thermocouple Types E, J, K and T:
± 0.01°C/°C ± 0.001% reading/°C.
Thermocouple Types R & S:
± 0.04°C/°C ± 0.001% reading/°C.

STABILITY WITH AMBIENT TEMPERATURE:
20 mV, 200 mV, 2 V and thermocouple ranges:
Greater than 1000 Megohms.
20 V and 200 V ranges:
1 Megohm.

INPUT IMPEDANCE:
20° F to 110° F.

POWER:
105 VAC to 125 VAC 50/60 Hz.
180 watts to 250 watts for typical systems.
220 VAC operation is optional.

OPERATING TEMPERATURE:

THERMOCOUPLE CONFORMITY:
The conversion of thermocouple voltage to temperature units is done by a digital computing circuit. The maximum difference between that conversion and the 1971 revised National Bureau of Standards thermocouple tables is given for the standard thermocouple circuits.

<table>
<thead>
<tr>
<th>Model No.</th>
<th>Material</th>
<th>Type</th>
<th>Range</th>
<th>Conformity</th>
</tr>
</thead>
<tbody>
<tr>
<td>8011F or C</td>
<td>Chromel-Constantan</td>
<td>E</td>
<td>-200°F to 1850°F (−130°C to 1010°C)</td>
<td>0.1°F</td>
</tr>
<tr>
<td>8012F or C</td>
<td>Iron-Constantan</td>
<td>J</td>
<td>-250°F to 1600°F (−155°C to 870°C)</td>
<td>0.1°F</td>
</tr>
<tr>
<td>8013F or C</td>
<td>Chromel-Alumel</td>
<td>K</td>
<td>-150°F to 2500°F (−100°C to 1370°C)</td>
<td>0.1°F</td>
</tr>
<tr>
<td>8014F or C</td>
<td>Plat.-Plat.-13% Rh.</td>
<td>R</td>
<td>500°F to 3200°F (250°C to 1760°C)</td>
<td>0.1°F</td>
</tr>
<tr>
<td>8024F or C</td>
<td>Plat.-Plat.-13% Rh.</td>
<td>R</td>
<td>32°F to 3200°F (0°C to 1760°C)</td>
<td>0.2°F</td>
</tr>
<tr>
<td>8015F or C</td>
<td>Plat.-Plat.-10% Rh.</td>
<td>S</td>
<td>500°F to 3200°F (250°C to 1760°C)</td>
<td>0.1°F</td>
</tr>
<tr>
<td>8025F or C</td>
<td>Plat.-Plat.-10% Rh.</td>
<td>S</td>
<td>32°F to 3200°F (0°C to 1760°C)</td>
<td>0.2°F</td>
</tr>
<tr>
<td>8016F or C</td>
<td>Copper-Constantan</td>
<td>T</td>
<td>-250°F to 800°F (−155°C to 242°C)</td>
<td>0.1°F</td>
</tr>
<tr>
<td>Feature</td>
<td>Standard</td>
<td>Optional</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-------------------------------</td>
<td>-------------------</td>
<td>---------------------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tape Width &amp; Thickness</td>
<td>1/2 in. wide, 1.5 mil thick</td>
<td>1/2 in. wide, 1.5 mil thick</td>
<td></td>
<td></td>
</tr>
<tr>
<td>DC Motor and position servo</td>
<td>DC motor and position servo</td>
<td>DC motor with velocity and position servo</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Power</td>
<td>115 VAC, 50-60 Hz</td>
<td>115 VAC, 50-60 Hz</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Feature</th>
<th>Standard</th>
<th>Optional</th>
</tr>
</thead>
<tbody>
<tr>
<td>Incremental Value (fps)</td>
<td>0-1000</td>
<td>800, (1000 Model, 1500 only)</td>
</tr>
<tr>
<td>Density (bps)</td>
<td>500</td>
<td></td>
</tr>
<tr>
<td>Bit Spacing Accuracy (using absolute, optical encoder)</td>
<td>12-1/2</td>
<td>1 to 25</td>
</tr>
<tr>
<td>Continuous Tape Speed (fps)</td>
<td>18, long-term</td>
<td></td>
</tr>
<tr>
<td>Speed Accuracy</td>
<td>18, instantaneous</td>
<td></td>
</tr>
<tr>
<td>Tape-Record Gap Time (ms)</td>
<td>30 and ≤ 104</td>
<td></td>
</tr>
<tr>
<td>Tape Gap Time (ms)</td>
<td>200</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: Specifications
SHADE RING TYPE 1 (DIFFUGRAPHY)

Essley pyranometer (180° pyrheliometer)
APPENDIX F

Experimental Data

Data Output Table Explanation

i.e., IC6078.307 - corresponds to test circuit no. 1

- Chamberlain collector
- collector tilt = 60°
- Year = 1978
- Julian Day No. 307

10:12:00 - 10:12:00 - starting time of data record in E.S.T.

Nomenclature

INT - Test interval - seconds
IT2 - Total solar radiation on collector surface, W/m²
IH2 - Total solar radiation on horizontal surface
IDF - Diffuse solar radiation on horizontal surface
TAM - Ambient air temperature °C
CII - Solar collector inlet fluid temperature
C01 - Solar collector outlet fluid temperature
WSP - Average wind speed m/sec.
WIDR - Wind Direction
FLR - Heat transfer fluid flowrate L/min.
(TI-TA)/I = (T_{fi} - T_{a})/I

EFF - Experimentally determined collector thermal efficiency
APPENDIX G

Frequency Analysis of Experimental Data

AXIS Label Description

\[ DT_1/I = (T_{T1} - T_{T2})/I \]

\[ WSP \equiv \text{wind speed in m/sec} \]

\[ FLR \equiv \text{heat transfer fluid flowrate, L/min} \]

\[ TAM \equiv \text{ambient air temperature} \]

\[ IT2 \equiv \text{Total Incident Solar Radiation on Collector Surface, W/m}^2 \]

\[ IDF \equiv \text{Horizontal Diffuse Radiation} \]

\[ DNN \equiv \text{Incident Angle of Beam Solar Radiation} \]
PLOT OF SOLAR COLLECTOR EFFICIENCY

Plot of WSP vs DTi/I
Plot of DNN vs DT1/I
APPENDIX H

Uncertainty Analysis for Experimental Determination
of Thermal Efficiency
APPENDIX H

Uncertainty Analysis for Experimental Determination of Thermal Efficiency

A formalized procedure for testing the thermal performance of solar collectors has been produced (4). This standards outlines the facility to be used, the accuracy and precision of the test instrumentation and the procedures required during testing.

The experimental equipment for this study was designed and chosen to meet the general requirements of this standard test procedure. In analyzing the test data an estimate of the uncertainty of the experimentally derived values of the collector thermal efficiency is required.

In performing any experiment an attempt is made to minimize the error associated with the experimental measurements. Unfortunately every measurement is subject to error and the degree to which this error is minimized is a compromise between the accuracy required in the derived value and the cost in money and time required to reduce the error in the component measurements to an acceptable value.

As a worst case, the error in a derived result is equal to sum of the errors of the component measurements. Implicit with this assumption is that the worst case errors will occur simultaneously and in the most detrimental fashion.

This condition is unlikely and consequently, usually overpredicts the error in a derived result. The odds of this condition occurring are small.
The concept of uncertainty implies random deviation about a mean value and subsequently some errors would be positive while others would be negative. Therefore, a series of measurements with relatively large uncertainties could produce a result with an uncertainty not much larger than the uncertainty of the most uncertain measurement (75).

The uncertainty in each component measurement may be described by the best estimate of the value for the variable, followed by the uncertainty interval at the desired confidence level (75) i.e.;

\[ V = M \pm W; \ P \ \text{percent} \]

where \( V \) is the viable
\( M \) is the best value
\( W \) is the uncertainty
\( P \) is the confidence level

Errors which introduce uncertainty into a derived result may be classified into two categories.

Systematic error – errors which persist and cannot be considered as due entirely to chance. Systematic errors may be due to incorrect instrument calibrations and relate to instrument accuracy, (the ability of the instrument to indicate the true value).

Random Error – errors which cause readings to take random values on either side of some mean value. An instrument which can reproduce subsequent readings
closely is said to have high precision and less random error.

Measurements may be highly precise but be inaccurate. During the commissioning of the experimental apparatus used for this study, the measurement equipment was calibrated against traceable standard instruments to minimize systematic errors. The basis of this thesis is that random errors that result in scatter in the experimental data are of major consideration.

In reviewing the uncertainty in an experimentally derived result, Kline and McClintock (76) have presented the following analysis.

Experiments fall into two overlapping categories - single-sample and multiple-sample experiments. Experiments in which uncertainties are evaluated by many repetitions and many diverse instruments are termed multiple-sample. Such experiments can be analyzed by classical statistical means.

The nature of the experiment described in section 7 is such that experimental conditions are subject to weather conditions that often occur infrequently during the year. This makes repetition of specific data virtually impossible.

The requirement to use fixed instrumentation to reduce scatter in the result due to systematic errors, precludes the second requirement of a multiple-sample experiment.

Experiments of the type described in section (7) which are characterized by small data sample sizes are termed single-sample experiments (76).
Since for single-sample experiments classical statistics cannot be used, an estimate of the reliability of the data must be based on estimates. The term "uncertainty" is defined as meaning a possible value the error might have (76). The propagation of uncertainty is defined as the way in which uncertainties in the variables affect the uncertainty in the results.

Kline and McClintock (76) have presented a technique for determining the propagation of errors in a derived result for single-sample experiments. In this technique a best estimate of the uncertainty in each variable is made by the researcher assuming odds (i.e., 20 to 1) that a measured value will fall within the uncertainty interval. If the same uncertainty estimate is used for all the component variables used in the analysis, the uncertainty in the result with the same uncertainty estimate will be given by:

\[ W_R = \left[ \left( \frac{\partial R}{\partial V_1} W_1 \right)^2 + \left( \frac{\partial R}{\partial V_2} W_2 \right)^2 + \ldots + \left( \frac{\partial R}{\partial V_n} W_n \right)^2 \right]^{\frac{1}{2}} \]

which may be reduced to:

\[ \frac{W_R}{R} = \left[ \left( \frac{W_1}{V_1} \right)^2 + \left( \frac{W_2}{V_2} \right)^2 + \ldots + \left( \frac{W_n}{V_n} \right)^2 \right]^{\frac{1}{2}} \]

In the experiments described in section 7 of this report, the experimental thermal efficiency of the Chamberlain solar collector is determined by the relationship,

\[ \eta = \frac{Q_u}{I_A c} - \frac{Q_{RHS}}{I_A c} \cdot \frac{\Delta T_c}{\Delta T_{RHS}} \]
Thus values of \( n \) are obtained by individual measurements of temperature rise across the solar collector and reference heater, measurements of reference heater power input and total solar radiation incident on the collector surface.

Power measurements to the reference heater were made with a Scientific Columbus, Inc. "Halltiplier" Watt Transducer, Model WT 5C5Al. The accuracy of this device is \( \pm 1\% \) of a full scale reading. The accuracy of the measurement of power input to the heat transfer fluid is estimated to be \( \pm 2\% \) overall. The measurement of solar radiation introduces the largest uncertainty and is estimated at \( \pm 5\% \) of reading. Errors in the measurement of temperature rise across the collector were \( \pm 0.1^\circ C \).

Therefore the uncertainty values are:

\[
\begin{align*}
in Q_{RHS} &= \frac{w_{Q_{RHS}}}{Q_{RHS}} = \pm 2\% \\
in I &= w_I = \pm 5\% \\
\Delta T &= w_{\Delta T} = \pm 0.1^\circ C
\end{align*}
\]

The values are based on best estimates of the values taking into account the manufacturer's calibration of the measurement equipment. The values represent the odds of 20 to 1 that measured values will fall within these limits. This may be representative of a confidence interval corresponds to 99.7\% of the values following within 3 \( \sigma \), (three standard deviations as defined in the classical statistical sense).
Applying the analysis (76), the uncertainty in determining \( n \) is:

\[
W_n = \pm \left[ \frac{W_{\text{RHS}}^2}{\Omega_{\text{RHS}}} + \left( \frac{W_I}{I} \right)^2 + \left( \frac{W_{\Delta T}}{\Delta T_{\text{RHS}}} \right)^2 + \left( \frac{W_{\Delta T}}{\Delta T_{\text{c}}} \right)^2 \right]^{\frac{1}{2}} \cdot \eta
\]

The uncertainty calculation is dominated by the uncertainty estimate in the radiation measurement \( I \). The value of \( W_n \) varies along the performance curve as a function of the value of the temperature rise across the collector. At high values of \( \eta \) the value of \( \Delta T_{\text{c}} \) is large and decreases linearly with decreasing \( \eta \). The error function is linear and smooth across the ranges of \((T_{f1} - T_a)/I\). The range of values observed is demonstrated below for the two limiting cases corresponding to \((T_{f1} - T_a)/I = 0 \) and \( 0.1 \)

For the case of \( \eta = 40\% \), \( \Delta T_{\text{c}} = 3.5^\circ\text{C} \)

\[
\Delta T_{\text{RHS}} = 6^\circ\text{C}
\]

Thus

\[
w_n = \pm \left[ (0.02)^2 + (0.05)^2 + (0.1)^2 + (0.1)^2 \right]^{\frac{1}{2}} \cdot 40
\]

\[
= 0.06 \quad (40) = \pm 2.4 \text{ Percentage Points}
\]

for the case \( \eta = 75\% \), \( \Delta T_{\text{c}} = 9^\circ\text{C} \)

\[
\Delta T_{\text{RHS}} = 6^\circ\text{C}
\]

\[
w_n = \pm \left[ (0.02)^2 + (0.05)^2 + (0.1)^2 + (0.1)^2 \right]^{\frac{1}{2}} \cdot 75
\]

\[
= 0.057 \quad (75) = \pm 4.3 \text{ Percentage Points}
\]

With an uncertainty estimate of 20 to 1, (30) that the measured values will fall within this range.
END

06-09-84

FIN