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FLAT PLATE COLLECTOR TESTING

Training in Alternative Energy Technologies

8TH SESSION

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Objectives

The objectives of the project were to:

(1) Acquire experience in conducting collector test using ASHRAE 93-77 as a guide

(2) Verify pertinent formulae and relationships

Introduction

As the energy crisis becomes more and more apparent due to political instability coupled with the drain in foreign exchange on the importation of oil the use of solar energy becomes a viable option. Solar-collectors being the devices for collecting solar energy become a critical factor in Solar Technology. There is the need to establish standards for the testing, construction and installation of solar collectors to insure good performance durability, reliability and safety.

Jamaica imports 90% of her energy requirements. The high cost of fuel coupled with unreliable supply have become a catalyst in the search for ways of reducing our dependency and consumption of conventional fuels. Considerable progress has been made through energy conservation and the development of alternative energy systems. Jamaica has a high level of insulation all year round and the use of solar thermal systems have a bright future. Courses are being conducted on design and intallation of solar thermal systems with the view to insuring proper standards and increasing production.

Even though Nigeria is an oil producing country, the use of free solar energy is very attractive. The utilization of solar energy will lead to reduction in local consumption of fuel and boost export resulting in more dollar power.

The desire to set up a standard for testing solar collectors motivated us to work on collector thermal efficiency testing.

Theory of Collector Testing

ASHRAE 93-77 method of collector testing was used as a guide. Not all areas were covered though; mainly because of time restrictions but our major objectives of being able to set up and conduct the experiment to obtain the pertinent data and do the necessary calculations were achieved.

Basic Efficiency Equations

The efficiency of a flat plate solar operating under steady state conditions may be described by the equation:

$$q/A_a = I_t(\tau\alpha) - U_L(T_p - T_a) \qquad (1)$$

F_R can be defined as the conventional heat exchanger effectiveness which is the ratio of the actual energy collected to maximum possible heat transfer. Introducting this factor in equation (1) results in:

$$q/A_a = F_p[I_t(\tau\alpha)_e - U_L(T_{fi} - T_a)$$
$$= \dot{m}C_p(T_{fe} - T_{fi})/A_a \qquad (2)$$

The solar collector efficiency is defined as:

$$\eta_g = \frac{\text{actual useful energy collected}}{\text{solar energy incident on collector}}$$

Equation (2) now becomes:

$$\eta_g = \left(\frac{A_a}{A_g}\right) F_R[(\tau\alpha)_e - U_L(T_{fi} - T_a) \qquad (3)$$
$$= \frac{\dot{m}C_p(T_{fe} - T_{fi})}{I_t A_g}$$

Equation (3) indicates that if η_g is plotted against $(T_{fi} - T_a)/I_t$, the result will be a straight line of slope equal to:

$\left(\frac{A_a}{A_g}\right) F_R' U_L$, and y intercept equal to: $\left(\frac{A_a}{A_g}\right) F_R(\tau\alpha)_e$

A straight line fit may not always suffice and it may be necessary to use a higher order fit, i.e. second order polynomial due to variation of U_L with temperature.

Collector Time Constant

It is necessary to define an equation that predicts the transient behavior of solar collector following a step change in incident radiation or inlet fluid temperature.

The governing equation for transient behavior is:

$$\frac{C_A}{A_a} \times \frac{dt_f}{dT} = F_r I_t (\tau\alpha)_e - F_r U_L (t_{fi} - t_a) - \frac{\dot{m}Cp}{A_a}(t_{f,e} - t_{f,i}) \qquad (4)$$

Solution of equation (4) yields:

$$\frac{F_R I_t (\tau\alpha)_e - F_R U_L (T_{f,i} - t_a) - (\dot{m}Cp/A_a)(t_{f,e,T} - t_{f,i})}{F_R I_t (\tau\alpha)_e - F_R U_L (T_{f,i} - t_a) - (\dot{m}Cp/Aa)(t_{f,e,initial} - t_{f,i})}$$

$$= e^{-[\dot{m}Cp/KC_A]T} \qquad (5)$$

$\frac{KC_A}{\dot{m}C}$ is the time constant. This is the time taken for the left hand side of equation (5) to change to 0.368 of its original value.

Collector Time Constant Computation

According to the test method, the incident solar radiation is equal to zero and $(t_{f,i} - t_a) = 0$

$$\frac{t_{f,e,T} - t_{f,i}}{t_{f,e,initial} - t_{f,i}} = 0.368 \qquad (6)$$

By monitoring entering and exit fluid temperatures as a function of time, the constant is the time required for the left hand side to be equal to 0.368. However, if the inlet fluid temperature cannot be controlled to within I 1°C(±1.8°F) it may be necessary to estimate $(A_a/A_g)\ F_R U_L$

for conditions of the test and time constant is now required for:

$$\frac{(A_a/A_g)F_R U_L(t_{f,i} - t_a) + \frac{\dot{m}Cp}{A_g}(t_{f,e,T} - t_{f,i})}{(A_a/A_g)F_R U_L(t_{f,i} - t_a) + \frac{\dot{m}Cp}{A_g}(t_{f,e,initial} - t_{f,i})} = 0.368 \qquad (7)$$

Angle of Incidence Modifier

The variation of $(\tau\alpha)$ with angle of incidence varies between collectors. It is necessary to determine the variation in incident angle and a factor $K_{\tau\alpha}$ called the incident angle modifier is introduced to effect this

Where $K_{\tau\alpha} = \frac{(\tau\alpha)_e}{(\tau\alpha)_{e,n}}$

Therefore equation (3) mow becomes:

$$\zeta_g = \frac{A_a}{A_g} F_R \left[K_{\tau\alpha}(\tau\alpha)_{e,n} - U_L \frac{(t_{f,i} - t_a)}{I_t}\right. \qquad (8)$$

A general equation for the variation of $K_{\tau\alpha}$ with Θ is given by:

$$K_{\tau\alpha} = 1 - b_o \left(\frac{1}{\cos\Theta} - 1\right) \qquad (9)$$

Collector Incident Angle Modifier Calculations

Three (3) different values of thermal efficiency shall be determined, corresponding to three (3) different values of incident angle. The differential between inlet fluid temperature and ambient is approximately zero and therefore, equation (3) reduces to:

$$K_{\tau\alpha} = \frac{\eta_g}{A_a/A_g \; F_R(\alpha\tau)_{e,n}} \qquad (10)$$

Three (3) different values of $K_{\tau\alpha}$ can be determined for the difference incident angles since the values of $(A_a/A_g)F_R(\tau\alpha)_{e,n}$ is known from the

intercept on the y axis of the efficiency curve. The value of b_o can be determined using equation (9) and the standard technique of least squares fit to a first order polynomial.

If $(t_{f,i} - t_a)$ cannot be controlled to within 11°C(11.8°F). them an estimate of $(A_a/A_g)\ F_R U_L$ os done for the collector for condition of test and each value of $K_{\tau\alpha}$ determined as:

$$K_{\tau\alpha} = \frac{\eta_g + (A_a/A_g)F_R U_L (t_{f,i} - t_a)/I}{(A_a/A_g)F_R(\tau\alpha)_{e,n}} \qquad (11)$$

The value of I used in equation (11) should be I_t for collectors that accept different fuse radiation and I_D for collectors that cannot accept diffuse radiation.

If the $(t_{f,i} - t_a)$ value cannot be made to be within $\pm$ 1°C ($\pm$1.8°F) then another method can be used which excludes the use of equation (11). This is done by plotting each data point on same plot with efficiency curve. A curve is drawn through each point parallel to the efficiency curve. The values of the y intercept are the values of efficiency that would occurred, had the desired temperature differential between inlet fluid and ambient be achieved. With these efficiency values the values $K_{\tau\alpha}$ can be determined.

Test Procedure Followed

In accordance with ASHRAE 93-77 a laboratory test was done on a locally available flat plate collector.

After the testing system was set up, water of inlet temperature 24.1°C at a constant mass flow rate of 0.0386 kg/s was made to pass through the collector. When thermal equilibrium was reached, all important data were taken.

This procedure was repeated for inlet temperatures of 28.2, 37.7, 43.5 and 51.6°C respectively at the same mass flow rate.

Observations

Conditions	(T) Temperature °C		
	T_{in}	T_{out}	T_{amb}
1	24.1	29.4	11.7
2	28.2	33.1	12.1
3	37.7	42.4	12.5
4	43.5	47.2	11.8
5	51.6	54.1	12.3

Calculations

Sample calculation of efficiency (ξ)

$$\eta_1 = (\dot{m}\, C_p\, \Delta T)/IA$$

but $\dot{m} = 0.0382$ kg/s

$C_p = 4190$ J/kgK

$\Delta T = T_o - T_{in}$

$= (29.8 - 24) = 5.8°$

$$\therefore \quad \eta_1 = \frac{(0.039 \times 4190 \times 5.8)}{704 \times 2.973} = 0.443 \text{ or } 44.3\%$$

Method Used to Determine Theoretical Heat Loss Coefficient (U_L)

In our determination of U_L we

(1) Fixed an average plate temperature

(2) Assumed a glazing temperature T_1

(3) Calculated heat transfer coefficients based on these temperature at local wind velocity at the average ambient temperature in the region of the collector

(4) Recalculate T_1 using $T_1 = T_p - \frac{U_t(T_p - T_a)}{h_{cp} + h_{rp}}$ based on the fixed plate temperature.

(5) Stop if recalculated T_1 is not significantly different from T_1 used in calculations. If not, repeat steps (3) and (4).

We verified by assuming different values of plate temperature (T_p) that U_L hardly varies with temperature and so an arbitrary assumption of a possible typical value of T_p does not appreciably affect the value of U_L.

Data Required for Theoretical Calculations

A_e	Area of Edge	=	$0.783\ m^2$
A_c	Area of Collector	=	$2.973\ m^2$
β	Collector Tilt	=	$0°$
d	Tube internal diameter	=	9.52×10^{-3} m
ε_1	Glass emmitance	=	0.85
ε_p	Plate Emmitance	=	0.92
σ	Boltzmann's Constant	=	$5.67 \times 10^{-8}\ w/m^2k^4$
δ	Absorber Plate thickness	=	1/6" = 1.159×10^{-3} m
K_p	Plate Conductivity	=	385 w/mk
K_i	Insulation Conductivity	=	0.16 wm/k
K_w	Conductivity of Water	=	0.654 w/mk
$\dot{m}$	Mass Flow Rate	=	$0.013\ kg/m^2s$
	Number of Tubes	=	6
Q	Volume Flow Rate	=	$0.013 \times 10^{-3}\ m^3/m^2/s$
W	Tube Spacing	=	0.20 m
Δx	Back Insulation Thickness	=	0.051 m
	Collector Dimensions	=	8' x 4' x 4"
	Distance Between Absorber and Glazing	=	1 3/4" = 44.5×10^{-3} m
	Bond Resistance	=	0 mk/w
τ	Transmittance of Glass	=	0.89
α	Absorptance of Absorber Plate	=	0.92

SAMPLE CALCULATIONS TO DETERMINE THEORETICAL EFFICIENCY OF COLLECTOR

Aim: The aim of these calculations is to compare theoretical and experimentally determined collector thermal efficiencies.

Absorber Plate to Cover

(i) Radiation $h_{rp} = \dfrac{\sigma\,(T_p^2 + T_1^2)(T_p + T_1)}{1/\varepsilon_p + 1/\varepsilon_1 - 1}$

$$h_{rp} = \frac{5.76 \times 10^{-8}\,(307 + 290)(307 + 290)}{\frac{1}{0.92} + \frac{1}{0.85} - 1}$$

$$h_{rp} = 4.79\ W/m^2\ K,\ 4.972$$

(ii) $h_{cp} = \dfrac{k}{d}\ Na\ \ W/m^2\ K$

$$Na = 1 + 1.44\,[1 - z]^+\ \ [1 - z(\sin 1.8\beta)^{1.6}] + [0.664z^{-1/3} - 1]^+$$

$$z = \frac{1708}{R\cos\beta}\ ,\ \beta = 0^o$$

$$R = g\Delta T\ d^3p^2\ C_p/\mu T\ \text{(Rayleigh number)}$$

$$R = 1.539 \times 10^6\ \frac{\Delta\tau}{T}$$

$$R = 8.76 \times 10^4$$

$$z = 1.95 \times 10^{-2}$$

$$\underline{\underline{Nu = 3.88}}$$

$$h_{cp} = \frac{k}{d}\ Nu$$

k = thermal conductivity of air
= 0.2931 W/mk

$d = 44.5 \times 10^{-3}m$

$$\therefore\ h_{cp} = 2.55\ W/m^2K$$

OUTER COVER TO ENVIRONMENT

Radiation $h_{r1} = \varepsilon\sigma(T_1^2 + T_s^2)(T_1 + T_s)$

$\varepsilon = 0.85$

T_1 = cover temperature

T_s = sky temperature

$h_{r1} = 0.85\ (5.67 \times 10^{-8})(307 + 284.7)(307 + 290)$

$= 4.99\ W/m^2k$

$= 4.67$

Convection $h_{c1} = 4.5 + 2.9u\ \ W/m^2k$

$h_{c1} = 4.79\ \ W/m^2k$

∴ Resistance of Cover to Environment

$$= \frac{1}{h_{c1} + h_{r1}} = \frac{1}{4.79 + 4.99} = 0.1022\ m^2k/W = 10.22 \times 10^{-2}\ m^2k/W$$

Resistance to heat transfer between absorber plate and cover

$$= \frac{1}{h_{cp} + h_{rp}} = \frac{1}{2.55 + 4.79}\ m^2k/W$$

$$= 13.62 \times 10^{-2}\ m^2k/W$$

$$U_t = \frac{1}{10.22 \times 10^{-2} + 13.62 \times 10^{-2}} = 4.17\ W/m^2k$$

$$U_b = \frac{k}{x}\left(1 + \frac{A_e}{A_c}\right)$$

polyurethane, K = 0.16 W m/k

Area of Edge $= 2\ (1.29^2 + 2.438)\ 0.101$

$A_e = 0.783m^2$

$\Delta x = 2'' = 0.051$ m

$$U_b = \frac{0.16}{0.051} + \frac{\frac{0.16}{0.051}(0.783)}{2.973}$$

$U_b = 3.96\ W/m^2k$

$\therefore U_L = U_b + U_t$

$U_L = 3.96 + 4.17$

$U_L = 8.15\ W/m^2k$

Recalculation of cover temperature

$$T_1 = T_p - \frac{U_t\ (T_p - T_a)}{h_{cp} + h_{rp}}$$

$$= 34 - \frac{4.19\ (34 - 11.7)}{2.55 + 4.79}$$

$T_1 = 21.33^oC = 294.33k$

$T_1 = 21.99^oC = 294.98k$

$\therefore$ We will have to recalculate the various heat losses and heat transfer coefficients using this new value of glazing temperature.

The final results are shown.

Final Results

(i) Absorber Plate to Cover Radiation (i) $h_{rp} = 4.972\ W/m^2k$

(ii) $h_{cp} = 3.14\ W/m^2k$

(ii) Outer Cover to Environment

Radiation; $h_{r1} = 4.67\ W/m^2k$

$h_{c1} = 4.79\ W/m^2k$

Resistance of Cover to Environment

$$= \frac{1}{h_{c1} + h_{r1}} = \frac{1}{4.79 + 4.67} = 10.57 \times 10^{-2}\ m^2k/W$$

Resistance of Absorber Plate to Cover

$$= \frac{1}{h_{cp} + h_{rp}} = \frac{1}{4.972 + 3.14} = 12.33 \times 10^2\ m^2k/W$$

$U_t = 4.37\ W/m^2k$

$U_b = 3.96\ W/m^2k$

$U_L = 8.32\ W/m^2k$

Volume Flow Through Tubes, $Q - 0.013 \times 10^{-3} \quad m^3/m^2.s$

$$= 0.013 \times 2.97 \ l/s = 0.038 \ l/s$$

Let V = velocity of fluid through tubes

$$Q = VA$$

$$V = \frac{Q}{A} = \frac{0.038 \times 10^{-3} \ m^3/s}{6 \times (71.2 \times 10^{-6})}$$

$$V = 0.0904 \ m/s$$

(number of tubes = 6)

$$Re = \frac{PVD}{\mu} = 336$$

$\therefore$ Flow is Laminar

We therefore have the situation where the flow is Laminar and fully developed. Also, since heat transfer per unit length of tube wall is constant

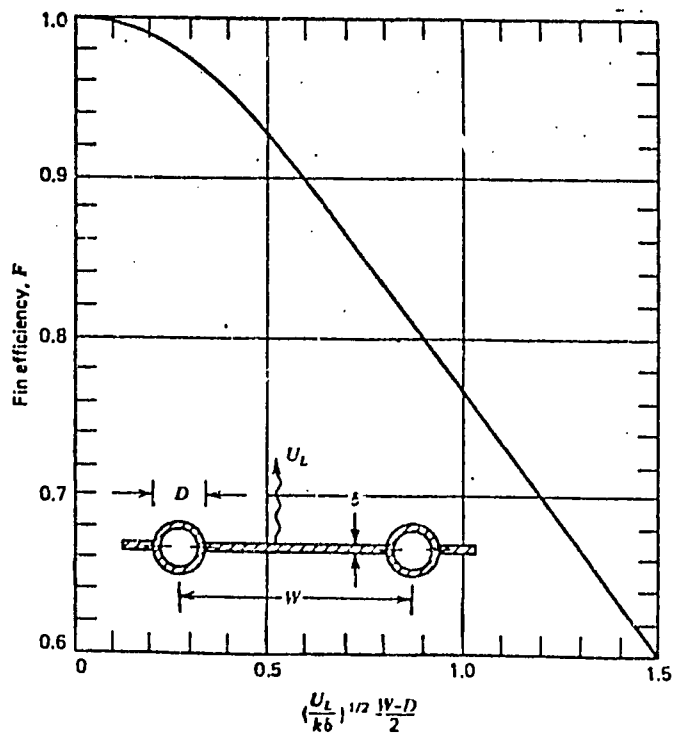
$$\underline{\underline{Nu = 4.364}}$$

$$h = Nu \frac{k_w}{D} = 4.364 \times \frac{0.654}{9.525 \times 10^{-3}}$$

$$h = 299.6 \ W/m^2k$$

$$m = \sqrt{U_L/k\delta}$$

$$m = \sqrt{\frac{8.32}{385 \times 1.59 \times 10^{-3}}} = 3.69$$



Fin efficiency for tube and sheet solar collectors.

$$F^1 = \frac{\frac{1}{U_L}}{W\left[\frac{1}{U_L\ [D + (W-D)\ F]}\right] + \frac{1}{C_B} + \frac{1}{\pi\ D_i\ h_f}\ ]}$$

$$= \frac{\frac{1}{8.32}}{0.2\ \left[\frac{1}{8.32\ [9.525 \times 10^{-3} + 0.19 \times .96]}\right. + 0 + \frac{1}{\pi \times 9.525 \times 10^{-3} \times 299.6}}$$

$$= \frac{6.777}{8.32} = 0.815$$

Dimensionless Collector Capacitance

$$c = \frac{\dot{m}\, C_p}{U_L\ F^1}$$

$$c = \frac{0.013 \times 4190}{2.973 \times 8.32 \times 0.815} = 2.70$$

$$F_R = \frac{\dot{m}\, C_p}{U_L} \left(1 - e^{-\frac{1}{c}}\right)$$

$$= \frac{0.013 \times 4190}{8.32 \times 2.973} \left(1 - e^{-\frac{1}{2.70}}\right)$$

$$\underline{F_R = 0.682}$$

With our values of F_R and U_L it is now possible to generate a theoretical efficiency curve.

U_L and F_R are both temperature dependent; however, not very much so and therefore we can safely assume constant values for these.

The efficiency of the collector operating under steady state conditions is given by

$$\eta = \left[F_R \tau\alpha - F_R\ U_L \left(\frac{Ti - Ta}{I}\right)\right]$$

We will assume various values of the fluid parameter $\left(\frac{Ti - Ta}{I}\right)$ from which we will calculate theoretical efficiencies

Let $\left(\frac{Ti - Ta}{I}\right)$ = 0.01, 0.02, 0.03, 0.04, 0.05, 0.06, 0.07

When $\frac{Ti - Ta}{I} = 0.01$

$$\eta = 0.682\ [0.89 \times 0.92 - 8.32\ (0.01)]$$

$$\eta = 0.46$$

Results

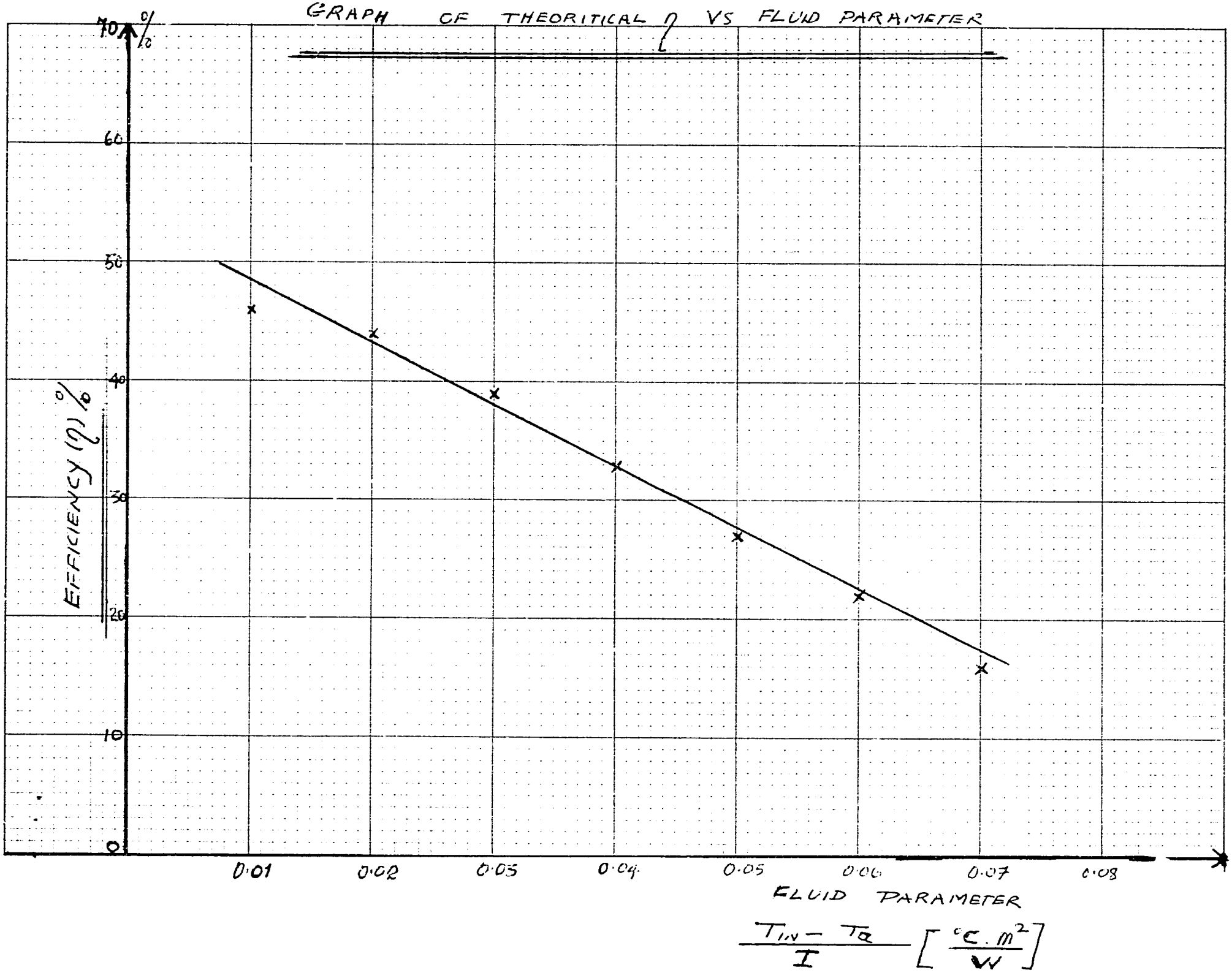
Theoritical Efficiencies vs Fluid Parameter

Fluid Parameter $\frac{T_i - T_a}{I}$	0.01	0.02	0.03	0.04	0.05	0.06	0.07
Efficiency (per unit)	0.46	0.44	0.39	0.33	0.27	0.22	0.16

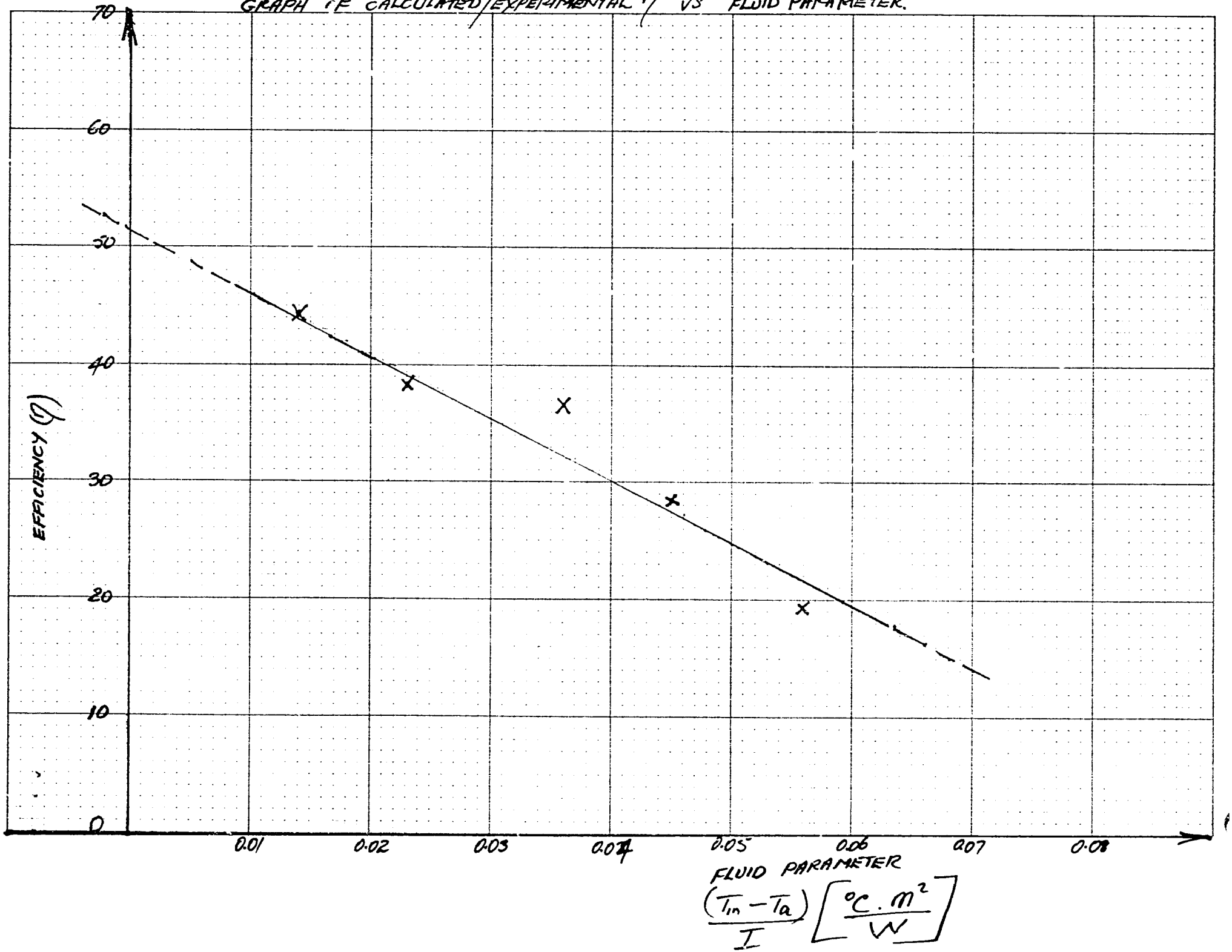
Experimental Efficiencies vs Fluid Parameter

Fluid Parameter $\frac{T_i - T_a}{I}$	0.014	0.023	0.036	0.045	0.056
Efficiency $(\eta)\%$	44.3	38.1	37.6	28.5	19.2

GRAPH OF THEORITICAL η VS FLUID PARAMETER



GRAPH OF CALCULATED/EXPERIMENTAL η 'VS' FLUID PARAMETER.



Discussion

A very important and interesting fact is that the theoretical efficiency vs $(T_{in} - T_A)/I$ may be generated from only the physical characteristics for any fluid flow rate.

With much success we were able to predict from theory the operating efficiency of the collector for a mass flow rate of 0.013 $kg/m^2/s$.

Two of the more important simplifying assumptions which might have some bearing on the theoretical results are:

(1) The physical characteristics such as τ, α and U_L are independent of temperature.

(2) The heat transfer from top and bottom of the collector are to to the same ambient temperature.

(3) There is no dust of other particles on collector glazing which might impair its transmissivity.

Theoretical $F_R U_L$ was 5.22 w/m^2k and actual $F_A U_L$ was 5.26 w/m^2k. However, in most cases the actual efficiency was lower than the theoretical efficiency. This implies that actual $F_R \tau\alpha$ was lower than theoretical $F_R \tau\alpha$.

This is probably due to a wrong estimate of $\tau\alpha$.

Advantages and Disadvantages of Laboratory Testing

There have been for some time now significant differences of view concerning the relative merits of testing solar systems/collectors in the field versus testing in the laboratory. There are many tasks which can be done more easily and more precisely in the laboratory while others are best done in the field. The decision as to which kind of

testing is best is contingent on the primary objectives for the specific case under consideration. Given below are some of the advantages and disadvantages of laboratory testing.

ADVANTAGES	DISADVANTAGES
(1) Parameters and variables are more easily controlled	(1) Results may not be representative of actual conditions
(2) Instruments may be more easily checked and recalibrated as necessary	(2) Typical test sequence unlikely to duplicate actual lifestyles
(3) Results are often reproduced	(3) Real world conditions not under test
(4) Component malfunction can be spotted early and corrected	(4) Common reasons for system failure or under performance may escape notice
(5) Data inaccuracies during acquisition are less likely	
(6) Weather data may be more precisely evaluated	
(7) Easies validation of computer models	
(8) Smaller samples can yield acceptable accuracy	

Short-Comings of the ASHRAE 93-77 Testing Procedure

The testing procedure recommended by ASHRAE 93-77 can only predict the instantaneous efficiency of a collector operating under the testing conditions. The experience is that the collectors behave differently from the predictions of ASHRAE 93-77 when they are in real world conditions.

There is the need for a more realistic and meaningful testing procedure, especially in the developing countries.

Concluding Remarks

Within the limits of experimental errors, the objectives of the project were realized.

References

(1) ASHRAE Standard 93-77

(2) Selected Topics in Alternative Energy Technologies for Developing Countries, 2nd Edition by Martin Bush, University of Florida

(3) ASHRAE Handbook 1977 Fundamentals

(4) Design of Solar Hot Water Systems for Jamaica - Prepared for Ministry of Mining & Energy USAID/GOJ Energy Assistance Project Charles J. Cromer, P.E.

(5) Fundamentals of Solar Energy Conversion 1983, Edward E. Anderson

NOMENCLATURE

a,b	= constants used in incident angle modifier equation, dimensionless
A	= cross-sectional area, $m^2(ft^2)$
A_a	= transparent frontal area for a flat-plate collector or aperture area for a concentrating collector, $m^2(ft^2)$
A_g	= gross collector area, $m^2(ft^2)$
A_n	= area of nozzle, $m^2(ft^2)$
A_r	= absorbing area of a flat-plate collector or the receiving area of a concentrating collector, $m^2(ft^2)$
b_o	= constant used in incident angle modifier equation, dimensionless
C_A	= effective heat capacity of the solar collector, J/°C(Btu/F)
C_n	= nozzle coefficient of discharge, dimensionless
c_p	= specific heat of the transfer fluid, J/(kg·°C) (Btu/(lbm·F))
D_n	= nozzle throat diameter, m(ft)
f_t	= temperature factor for calculation of nozzle Reynolds number, dimensionless
F′	= absorber plate efficiency factor, dimensionless
F_R	= solar collector heat removal factor, dimensionless
I	= solar irradiation, $W/m^2(Btu/(hr \cdot ft^2))$
I_D	= direct solar irradiation component $W/m^2(Btu/(hr \cdot ft^2))$
I_{DN}	= direct normal solar irradiation, $W/m^2(Btu/(hr \cdot ft^2))$
I_d	= diffuse solar irradiation incident upon the aperture plane of collector, $W/m^2(Btu/(hr \cdot ft^2))$
I_{sc}	= solar constant, 1 353 W/m^2(429.2 $Btu/(hr \cdot ft^2)$)
I_t	= total solar irradiation incident upon the aperture plane of collector, $W/m^2(Btu/(hr \cdot ft^2))$
K	= factor defined by equation (8.7); dimensionless
$K_{\alpha\tau}$	= incident angle modifier, dimensionless
LST	= local standard time, decimal hours
LSTM	= local standard time meridian, deg
AST	= apparent solar time, decimal hours
m	= air mass, dimensionless
$\dot{m}$	= mass flow rate of the transfer fluid, kg/s (lbm/hr)
N_{Re}	= Reynolds number, dimensionless
P_{th}	= theoretical power required to move the transfer fluid through the collector, W(hp)
p_n	= absolute pressure at the nozzle throat, Pa(lbf/in.^{2})
p_v	= velocity pressure at the nozzle throat or the static pressure difference across the nozzle, Pa(lbf/in.^{2})
Δp	= pressure drop across the collector, Pa(lbf/in.^{2})
Δp_n	= pressure drop across the nozzle, Pa(lbf/in.^{2})
Q_{mi}	= measured air flow rate, $m^3/s(ft^3/min)$
Q_s	= standard air flow rate, $m^3/s(ft^3/min)$
q_u	= rate of useful energy extraction from the collector, W(Btu/hr)
t_a	= ambient air temperature, °C(F)
t_f	= average fluid temperature, °C(F)
$t_{f,e}$	= temperature of the transfer fluid leaving the collector, °C(F)
$t_{f,e,initial}$	= temperature of transfer fluid leaving the collector at the beginning of a specified time period, °C(F)
$t_{f,e,T}$	= temperature of the transfer fluid leaving the collector at a specified time, °C(F)
$t_{f,i}$	= temperature of the transfer fluid entering the collector, °C(F)
t_p	= average temperature of the absorbing surface for a flat-plate collector, °C(F)
t_r	= average temperature of the absorbing surface for a concentrating collector, °C(F)
Δt	= temperature difference, °C(F)
U_L	= solar collector heat transfer loss coefficient, $W/(m^2 \cdot °C)$ $(Btu/(hr \cdot ft^2 \cdot F))$
V_a	= velocity of the air at the nozzle throat, m/s (ft/min)
v_n	= specific volume of the air at the nozzle at standard barometric pressure, m^3/kg dry air (ft^3/lbm dry air)
v_n'	= specific volume of air at the nozzle per unit mass of air-water vapor mixture, m^3/kg (ft^3/lbm)
w	= density, kg/m^3 (lbm/ft^3)
W_n	= humidity ratio at the nozzle, kg H_2O/kg dry air (lbm H_2O/lbm dry air)
α	= absorptance of the collector absorber surface for solar radiation
Γ	= fraction of specularly reflected radiation from the reflector or refracted radiation which is intercepted by the solar collector absorbing surface
γ	= collector-solar azimuth, deg
θ	= angle of incidence between direct solar rays and the normal to the collector surface or to the aperture, deg
β	= solar altitude angle, deg
ϕ	= solar azimuth angle, deg
ψ	= collector azimuth angle, (measured from the south in the horizontal plane), deg
η_g	= collector efficiency based on gross collector area, %
λ	= wavelength, μ m
ϱ	= specular reflectance of the solar collector reflector
T	= time, decimal hours or seconds
τ	= transmittance of the solar collector cover plate, dimensionless (if no cover plate is used, τ=1.0)
$(\tau\alpha)_e$	= effective transmittance-absorptance product, dimensionless
$(\tau\alpha)_{e,n}$	= effective transmittance-absorptance product at normal incidence
T_1, T_2	= time at the beginning and end of a test period, decimal hours
Σ	= collector tilt from the horizontal, deg