# THE POTENTIAL USE OF SOLAR ENERGY
# FOR
# TOBACCO CURING IN THAILAND

Investigator: A. HIRUN
Department of Mechanical Engineering
Chiang Mai University
Chiang Mai, Thailand

Training in Alternative Energy Technologies Program
8th Session
December 1983
University of Florida
Gainesville, Florida
U.S.A.

## ABSTRACT

A. Hirun, 1983. The Potential Use of Solar Energy for Tobacco Curing in Thailand. TAET, USAID and University of Florida, Gainesville, Florida, USA.

Thai flue-cured tobacco is an important foreign exchange earner for Thailand. The wood-fired curing barns are very inefficient and as the result, over 1 million $m^3$ of firewood are being wasted annually. The investigation shows only 7% thermal efficiency for the Thai wood-fired barn. Two different flat-plate solar air heaters are tested and compared. The investigation shows how tin-cans can be used as the absorber for the solar air heater. The tests confirm that the tin-can collector is more efficient than the collector using black polyethelene film as the absorber. A temperature difference of upto 30°C across the collector surface is possible. The potential use of solar energy to implement some of the heat requirement for tobacco curing in Thailand appears logical. Further work is needed to assess the extent of this potential.

# ACKNOWLEDGEMENTS

This investigation is part of the 8th session of the Training in Alternative Energy Technologies (TAET) program. The investigator wishes to express his deep appreciation to several staff members of this program for their kind cooperation. Special thanks are due to Vann Chesney for his assistance in the workshop and, also, to Nancy Allen and Sharon Loschke for their hard work in helping prepare this report.

The investigator is deeply indepted to the USAID for providing the financial assistance and to the University of Florida for hosting this TAET program. Without them this investigation would not have been possible.

# Table of Contents

# List of Figures

# List of Tables

# PART 1 Overview of Tobacco Curing Industry in Thailand

## 1.1 Introduction

Tobacco is an important cash crop and a significant foreign exchange earner for Thailand. In 1982, Thailand's tobacco production amounted to 63,100 tonnes, of which about two-third are exported with a total value of US 109.8 millions.[1]

Thai tobacco is well known all over the world for its excellent characteristics, such as low tar, low nicotine and high sugar. Furthermore, its neutrality makes it very suitable for use in blended tobacco. Four varieties of tobacco are being grown in Thailand. These are Virginia, Turkish, Burley and a local variety. Of these four, Virginia tobacco is the most widely grown with its annual production amounting to about 80% of the total.

Tobacco leaves are cured immediately after harvest, usually in local tobacco curing stations. At present, there are approximately 400 stations, mostly situated in northern Thailand. Each station has, on the average, 20-25 curing barns, most of which use firewood as fuel. It is estimated that the 1982 energy consumption for tobacco curing in Thailand amounts to about 1 million $m^3$ of firewood equivalent.[2] This substantial amount has resulted in a large scale deforestation for the past two decades. The consequences of such deforestation are very serious and wide-ranging, as are now well known. Clearly, a reduction of firewood wasted in the tabacco curing industry will help alleviate the massive destruction of forest.

Northern Thailand is under high solar insolation during the months from December to April which coincides with the tobacco curing period. The potential use of solar energy to provide some of the heat requirement for tobacco curing in Thailand appears very bright. Work on solar tobacco curing systems in the United States and other tobacco producing countries has resulted with some optimism.

Note: It has been reported by Johnson (1980) that 11% of the total energy required in tobacco curing, using bulk curing, can be supplied by solar energy (North Carolina).[3] Even, larger energy reductions are possible, by incorporating energy conservation measures along with new technology such as solar collector and heat recovery. This has been accomplished by Cundiff (1980), where up to one-third of the required energy is supplied by solar energy.[4]

## 1.2 Tobacco Curing Processes

There are two tobacco curing methods being employed today by various tobacco growers around the world. The most common method is the classical technique known as flue curing. Whereas, the more recent one is by bulk curing. These two methods are illustrated in figures 1 and 2.

All of the existing curing barns in Thailand make use of flue curing. This term is derived from the process of drying or curing the leaves in which heat is distributed uniformly by the use of a heat exchanger of flue tubes inside a tightly closed barn. The leaves are loosely hung inside the barn so that easy flow of hot air is permitted. The mode of heat transfer to the leaves is mainly by natural convection. The fuel which may be wood, coal, LPG or fuel oil is burnt in the burner, and the hot flue gases flow through the metal tubes which act as a heat exchanger.

For bulk curing, the leaves are firmly packed, hung or placed in racks. Hot air is driven by fans or blowers through the leaves. The heat source may be from gas or oil fired burners.

The process of flue curing involves promoting the yellowing of the leaves and preventing them from becoming brown. This is done by controlling the temperature and relative humidity so that after about four days of gradual increases in temperature and the commensurate reduction in relative humidity, the final product is in the form of golden to yellowish dried leaves with 15 to 20 percent moisture content (wet basis).

The temperature and relative humidity requirements of the four stages in flue curing are summarized in Table 1 below.

| Stages | Time (Hr) | Temperature (°c) | RH (%) |
|---|---|---|---|
| 1. Yellowing | 24-36 | 32-43 | 80-90 |
| 2. Color fixing | 6-24 | 45-53 | 50-80 |
| 3. Leaf drying | 9-14 | 55-63 | 30-50 |
| 4. Stem drying | 17-25 | 65-80 | 20-30 |

Table 1 Summary of Requirements by Thailand Flue Cured Tobacco.[5]

Figure 1 Conventional Flue Curing Barn Utilizing Natural Air Ventilation

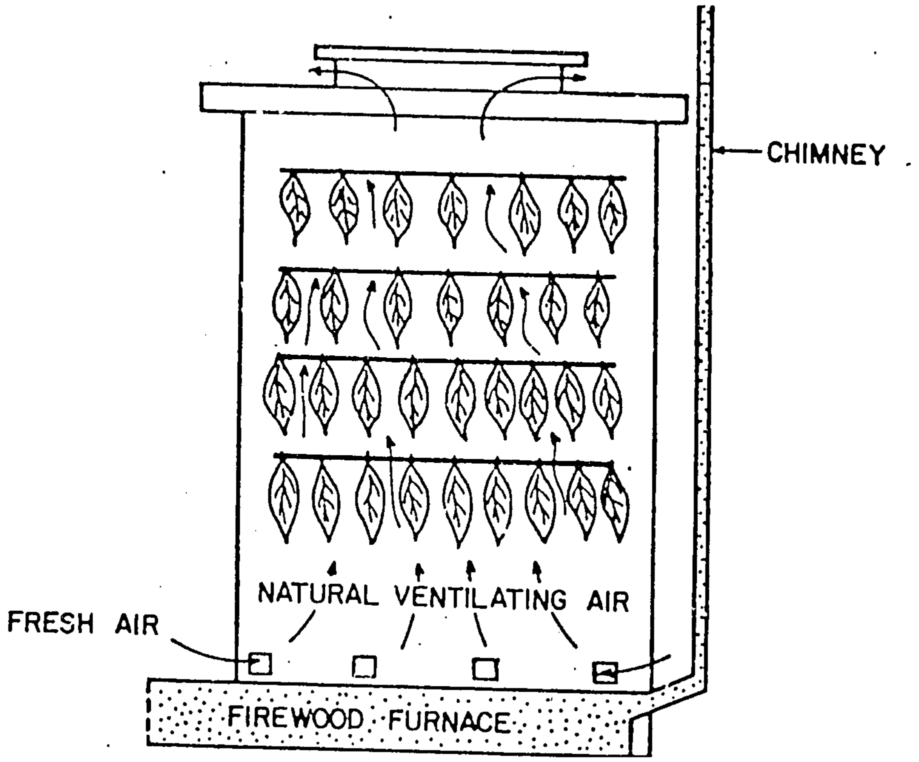
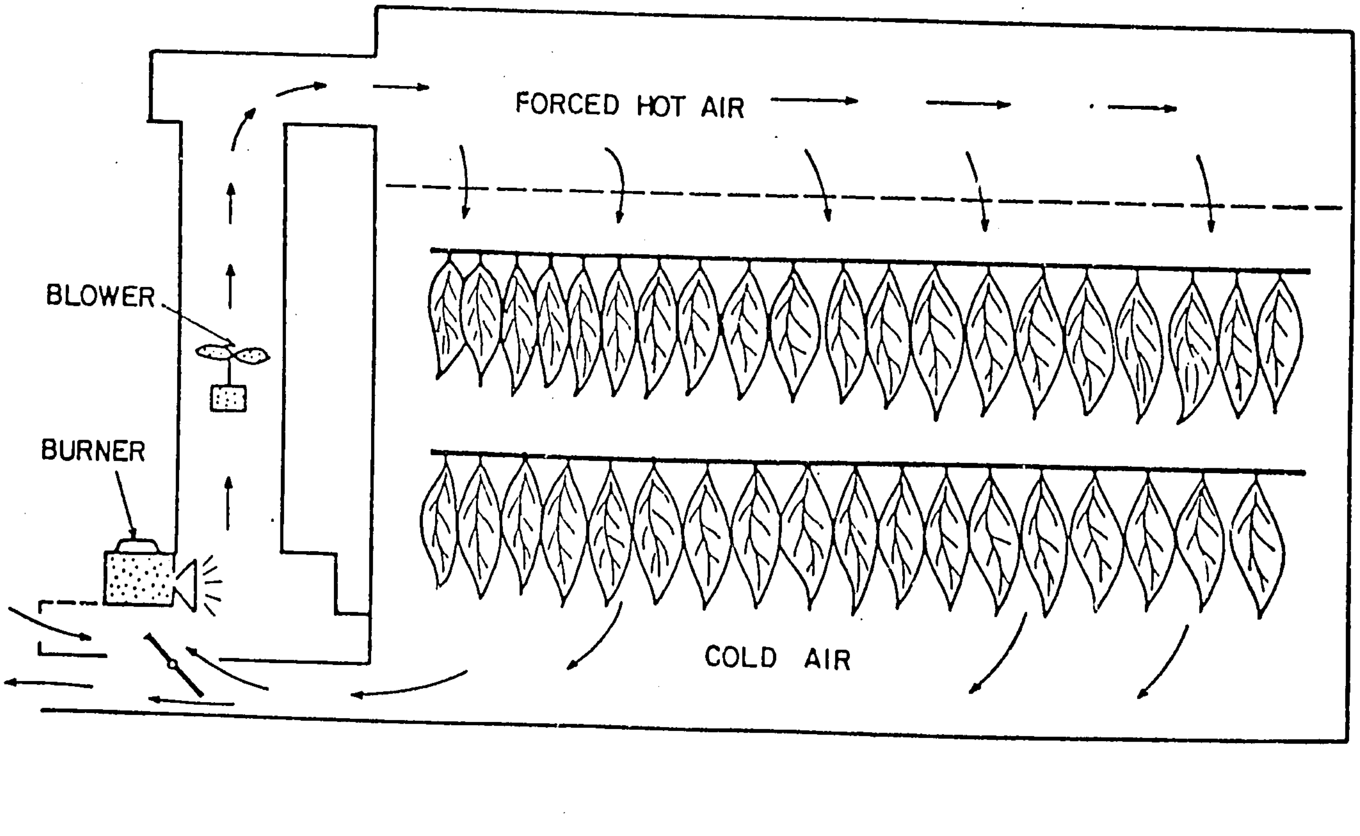


Figure 2 Bulk Curing Barn Using Forced Convection



## 1.3 Energy Usage in Curing of Tabacco Leaves in Thailand

The standard flue curing barns used in Thailand are usually made of bricks or hollow blocks for the four side walls and galvanized iron sheets for roofing material. The furnace is normally made of bricks and is connected to a metal heat exchanger. The barn's dimension is generally about 6x6 $m^2$ in floor area and 9 m in height. Each curing barn can cure approximately 1,500 kg of fresh tobacco leaves, at 80 to 85 percent moisture content (wet basis), in one curing batch.

Tobacco curing barns in Thailand mainly use wood as the energy source. In some province (counties), where deforestration is being checked, the curing operators are forced to use lignite as the energy source. Table 2 indicates the estimated total energy consumption by the tobacco-curing industry in Thailand, in terms of wood and lignite equivalent.

| | 1977 | 1978 | 1979 | 1980 | 1981 | 1982 |
|---|---|---|---|---|---|---|
| Annual Tobacco Production (tonnes) | 51,800 | 55,600 | 59,600 | 47,200 | 59,300 | 63,100 |
| Firewood-equivalent in Fuel ($m^3$) | 859,880 | 922,960 | 983,360 | 783,520 | 984,380 | 1047,460 |
| Lignite-equivalent in Fuel (tonnes) | 362,000 | 389,200 | 417,200 | 330,400 | 415,100 | 441,70 |

Table 2 Energy Consumption by Tobacco-Curing Industry in Thailand.[6]

NOTE: From Bamrungwong (1980), 1 kg of cured leaves requires 0.0166$m^3$ (120 MJ) of firewood or 7 kg (182 MJ) of lignite for energy source.[7]

## 1.4 Estimated Efficiency of Thai Curing Barnes

Thermal efficiency of the curing barns in Thailand has never been studied. However, based on the known energy requirement per kg of cured leaves and the energy input, estimation of the thermal efficiency for the existing curing barn can be made as follows.

Assumptions:

1 Tobacco leaves are harvested at 80% moisture content (wet basis) or 400% moisture content (dry basis).

2 The moisture content of the cured tobacco leaves is reduced to 20% moisture content (wet basis).

Definitions:

% Moisture (wet basis), $M_W = 100\, W_W/(W_W + W_D)$

% Moisture (dry basis), $M_D = 100\, W_W/W_D$

where, $W_W$ = mass of water.

$W_D$ = mass of dry matter.

Relationship for evaporation load:

$$m_w = m_i\,(M_i - M_f)\,/\,(100 - M_f)$$

where, $m_w$ = evaporation load (kg),

$m_i$ = initial mass of wet crop (kg.

$M_i$ = initial moisture content (%).

$M_f$ = final moisture content (%)

Since tobacco leaves will be cured from 80% to 20% moisture content (wet basis), and assuming fresh leaves of 1 kg, hence

$$m_w = \frac{1\,(80-20)}{100-20} = 0.75 \text{ kg of water/kg of fresh leaves.}$$

This means that in order to obtain 0.25 kg of cured leaves, 0.75 kg of water must be removed from the fresh leaves, or 3 kg of water must be removed to obtain 1 kg of cured leaves.

Assuming that water in the leaves is at 50°C, its latent heat of evaporation is about 2.4 MJ/kg. Therefore, to remove 3 kg of water requires 3 x 2.4 = 7.2 MJ of heat. However, if consideration is given to the fact that evaporation of water from agricultural produce requires more energy than from free surface water, and assuming this requires 15% more energy for tobacco leaves, hence, the energy required to evaporate 3 kg of water from tobacco leaves will be 1.15 x 7.2 MJ = 8.28 MJ.[8]

Efficiency of Thai drying barns,

$$= \frac{\text{evaporation enthalpy/kg of cured leaves}}{\text{energy input/kg of cured leaves}} \times 100$$

Firewood ease, $= \frac{8.28}{120} \times 100 = 6.9\%$

Lignite case, $= \frac{8.28}{182} \times 100 = 4.6\%$

From the above calculations, the thermal efficiency of curing barns in Thailand is 6.9% (for Firewood fuel) and 4.6% (for lignit fuel). Lignite fuel barns are surprisingly less efficient. The reasons for this are believed to be the uncontrolled combustion which results in losses of sensible heat through flue gases, poor quality of lignite from northern Thailand, and more importantly, the fact that lignite is used in the existing unmodified wood fuel barns.

It is important to note that the thermal efficiency of Thai curing barns is poor when compared with that of US curing barns which adopt a bulk curing technique. For example, Cundiff (1980) obtained a seasonal energy usage of only 18 MJ/kg of cured leaves from an insulated bulk curing barn using petroleum fuels.[9] In uninsulated commercial barns using average management, Watkins (1978) found that three LPG powered barns averaged 32 MJ/kg of cured leaves, and fuel oil-powered barns averaged 49 MJ/kg of cured leaves.[10]

# PART 2 Thermal Efficiency Test of Two Flat-plate Solar Air Heaters

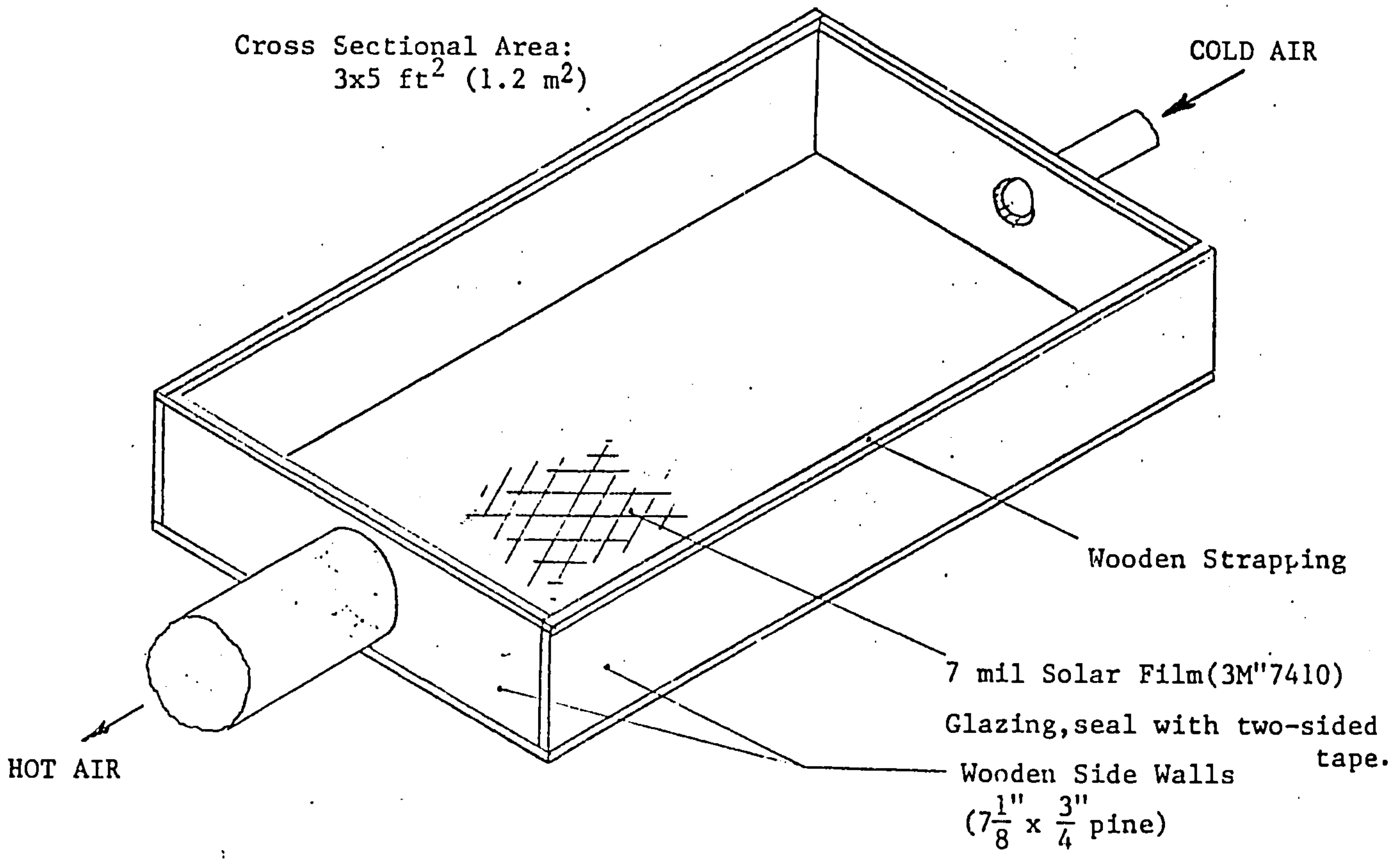
## 2.1 Background

The primary objective of this investigation is to test and determine the thermal performance of two flat-plate solar air heater modules (hereinafter called solar collectors). The thermal efficiency curve for each collector is to be compared with that of a compatible ASHRAE STANDARD 93-77 test.[11] The more efficient collector is to be further tested to determine thermal efficiency curves for a combination of values of air flow rate and solar simulator insolation.

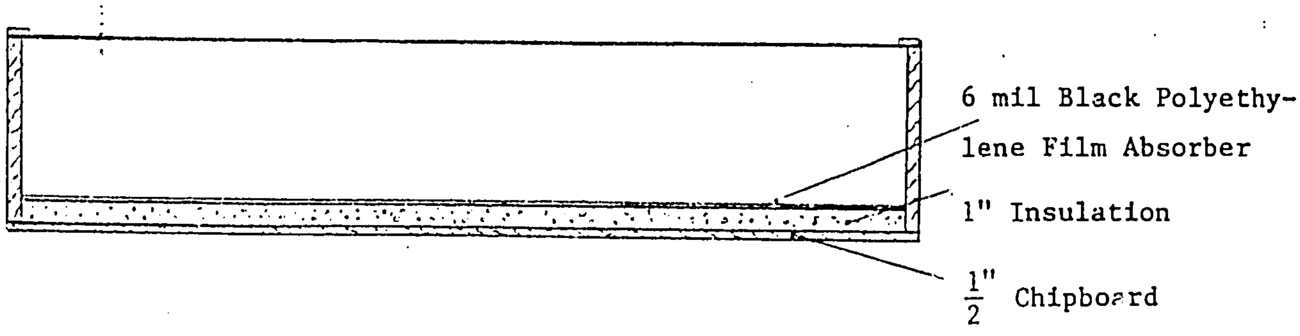
The two collectors to be tested are illustrated in Figure 3 and Figure 4. Both collectors are of similar configuration and material, except for the absorber plate. The first collector used black polyethylene film, as widely used in most solar crop dryers, for absorbing solar energy. The other collector used cut out tin-cans, painted black, as absorber. The purpose is to increase the temperature difference ($\Delta T$) across the collector by using discarded metal materials, so as to keep the cost of the collector down to a minimum.

The ASHRAE STANDARD 93-77 test provides standard procedures where solar energy collectors can be tested both indoors and outdoors, to rate the collectors in accordance with their thermal performance. The thermal performance of the solar collector is determined by obtaining values of instantaneous efficiency. This requires experimentally measuring the rate of incident solar radiation onto the solar collector as well as the rate of energy addition to the transfer fluid as it passes through the collector, at different inlet fluid temperature, all under steady state conditions. The tests to be performed will use air as the transfer fluid and a solar simulator will simulate the indoor sun.

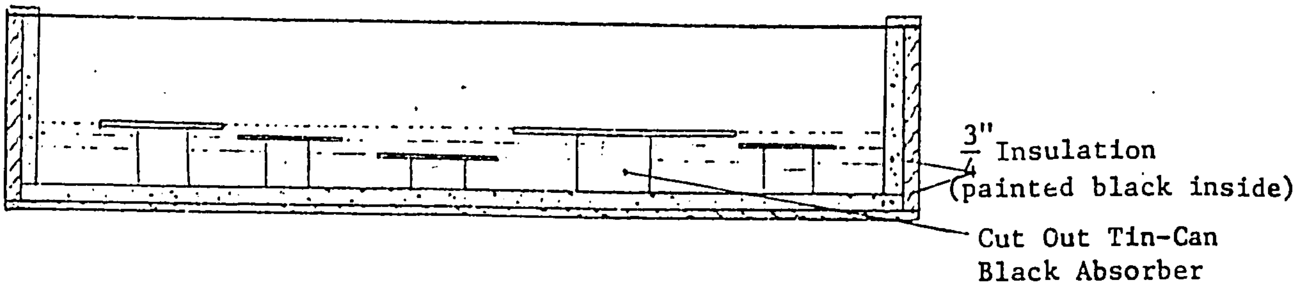
Figure 3 TEST MODEL SINGLE-GLAZED FLAT-PLATE AIR HEATERS



BLACK POLYETHYLENE FILM ABSORBER AIR HEATER Cross-Section



CUT OUT TIN-CAN ABSORBER AIR HEATER Cross-Section



Insulation: Polyisocyanurate Foam Board with Aluminium Foil Face (Celotex)

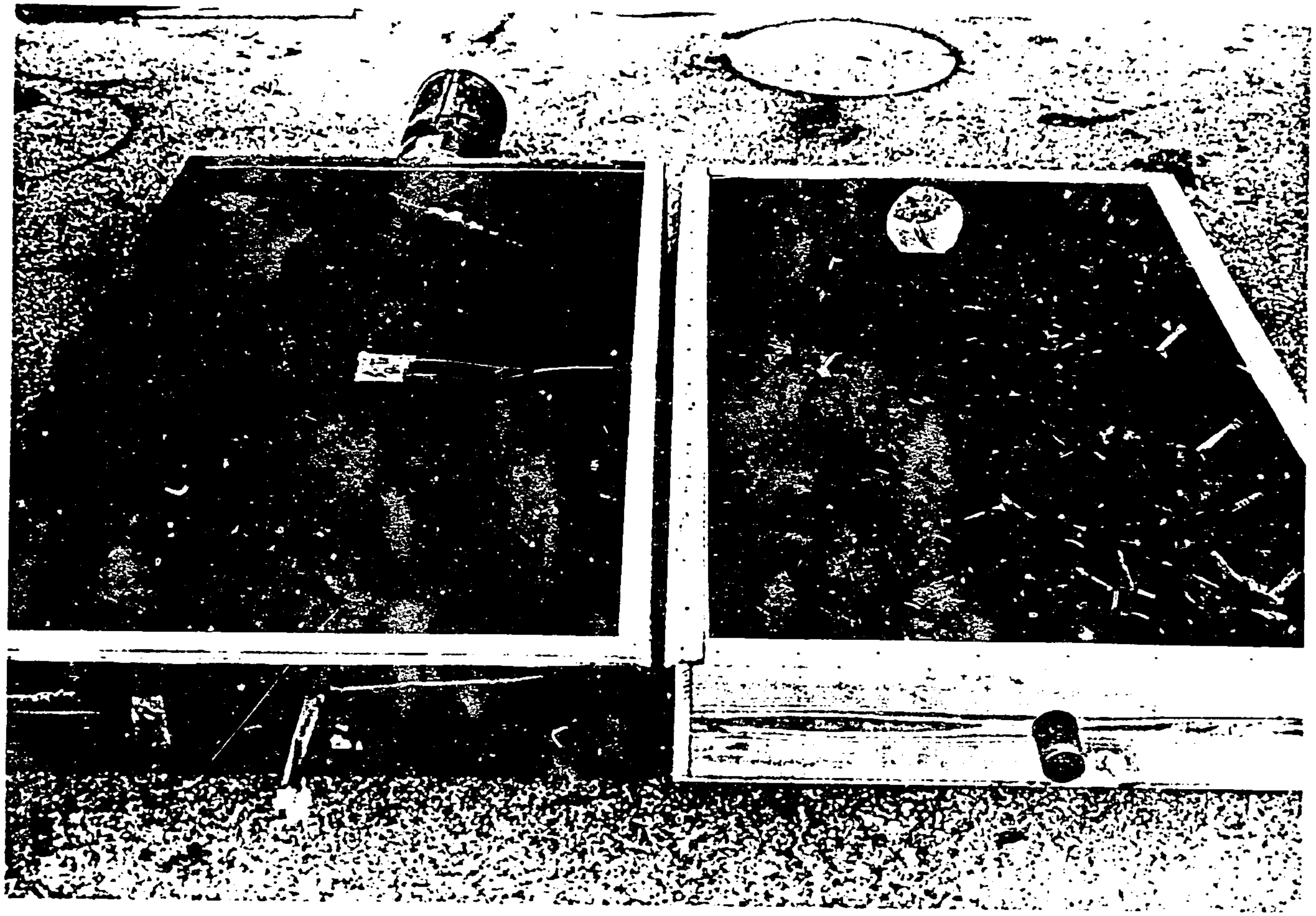


Figure 4 Photographic View of the Two Experimental Air Heaters

## 2.2 Hardware and Instrumentation

The hardware and instrumation items used in the thermal performance tests are set as shown in Figure 5 and Figure 6. Each item is labelled and used as follows.

1 Indoor Solar Simulation

The solar simulator consists of 6 x 6 x 150 W flood light bulbs mounted on a stand.

2 Insulation Measurement

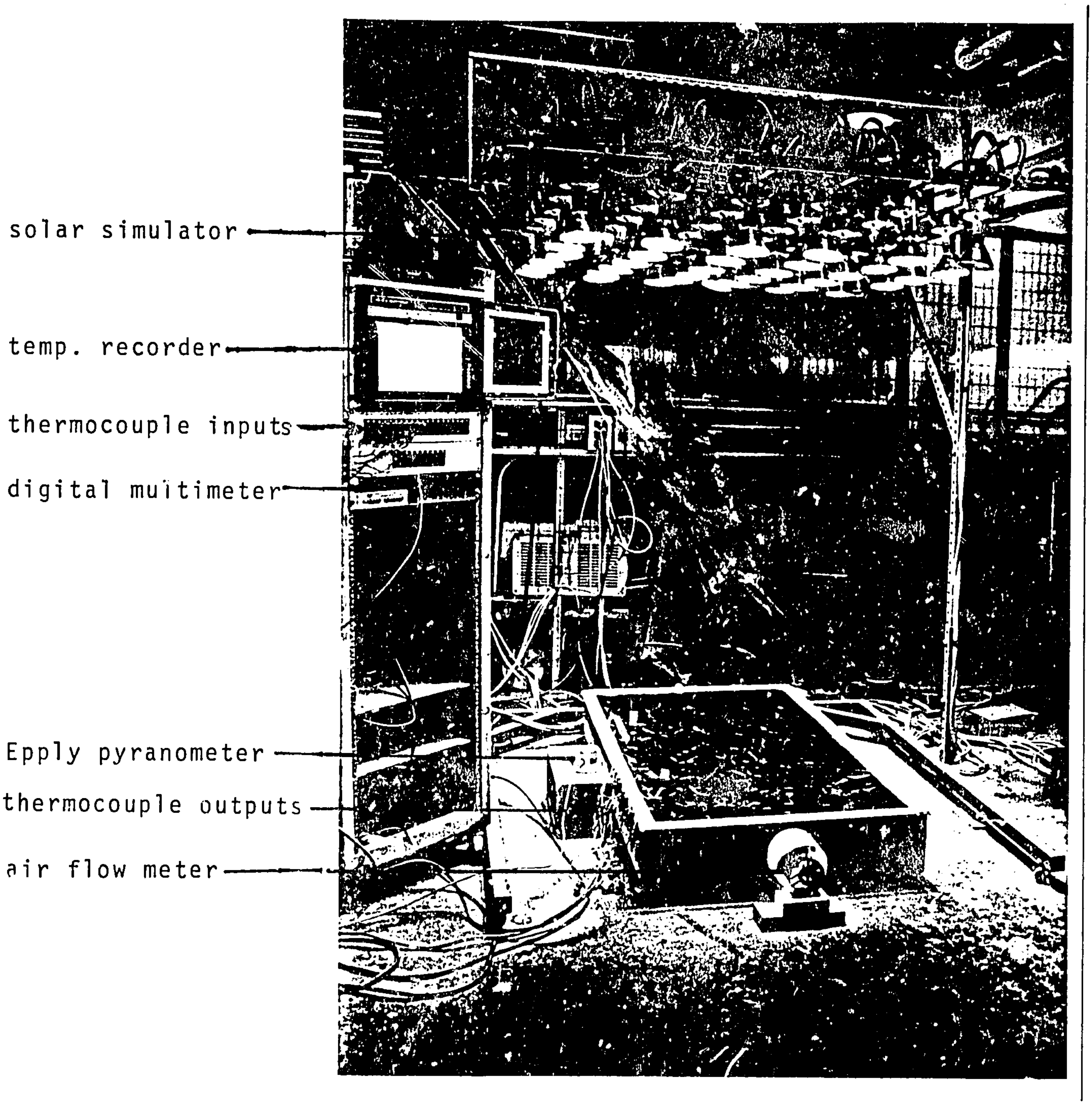
The indoor solar simulator insolation values are measured by an Epply white and black pyranometer model 8-48: SN 20160, with calibration constant K = 89.265 $w/(m^2\text{-}mv)$. A Hewlett - Packard 3465A digital multimeter is used to measure the milli-volts equivalent of the insolation values.

3 Temperature Measurments

The necessary temperature measurements are made by using OMEGA Type T (Copper-Constantan) thermocouple wire. Temperature difference of the transfer fluid across the solar collector ($T_o$-$T_i$) is monitored by HEWLETT - PACKARD 3465A digital multimeter. The same instrument is used to measure the temperature difference between the inlet and ambient air ($T_i$-$T_a$). Steady state conditions will be indicated by monitoring various temperature points across the collector with the use of a LEEDS and NORTHRUP SPEEDOMAX RECORDER T-68 (30 channels).

Temperature distribution and thermal performance of the cut out tin-can collector can be extensively analysed by incorporating the following nine thermocouple points:

Figure 5 Photographic View of the Hardware and Instrumentation Test Configuration



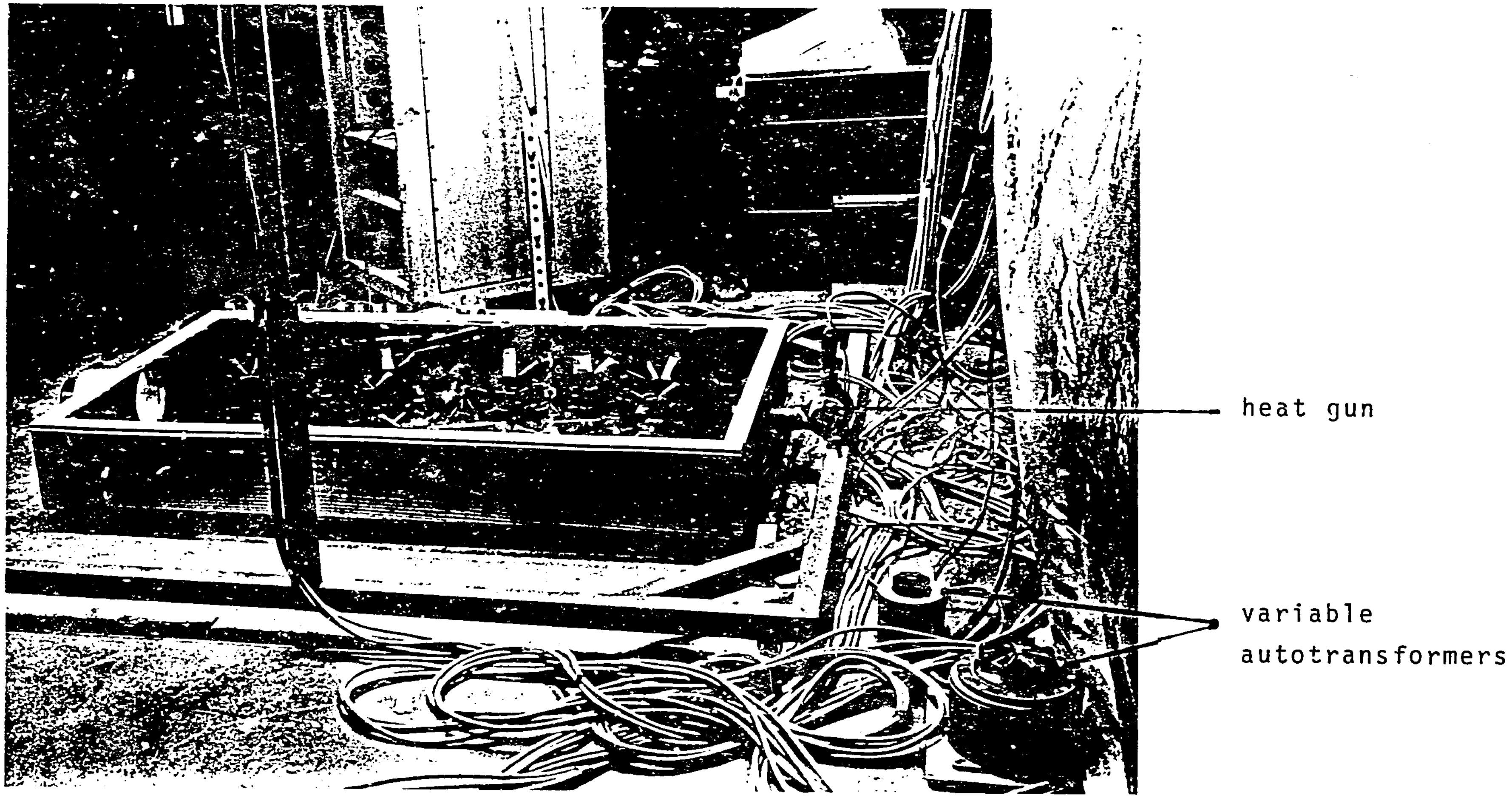


Figure 6 Photographic View of the Cut Out Tin-Can Air Heater with Air Inlet Hardware

- Ambient air temperature ($T_a$)
- Inlet air temperature ($T_i$)
- Air temperature at 1/4 length from inlet
- Air temperature at 1/2 length from inlet
- Air temperature at 3/4 length from inlet
- Outlet air temperature ($T_0$)
- Aluminium foil bottom plate temperature
- Aluminium fin (soft drink cans) temperature
- Steel tin-can fin temperature

4 Air Flow Measurement

The air flow rate is determined by the use of WM W 131 SN 645 air meter, a rotating vane type anemometer.

5 Inlet Air Adjustment

The combination of values of the inlet air flow rate and inlet temperature is achieved by using a heat gun and two variable autotransformers. The heat gun (115V x 20A max) heater and blower speed are separately controlled by the two variable autotransformers (0-120V, 22A max). For $T_i$ equals $T_a$ tests, another blower (115V, 0-21A, 3350RPM max) is used to achieve higher air flow rates.

NOTE: The equipments selected are readily available in the TAET laboratory and their details are fully explained in the "Solar and Ancillary Measurement Equipments" handbook.[12]

## 2.3 Test Procedures and Computations

The test configurations for testing the two solar air heaters are shown in Figures 5 and Figure 6. There are two parts of the test to be preformed. The first is to determine the thermal efficiency of each collector for varying air inlet temperatures, and tested under two constant sets of flow rates. The second is to further test the more efficient collector by obtaining values of instantances efficiency for a combination of values of insolation and flow rate, but under $T_i$ equals $T_a$ condition.

The performance of a flat-plate solar collector operationg under steady state conditions can be described by the following relationship:

$$q_u/A_a = I(\tau\alpha)_e - U_L\ (t_p - t_a) = \dot{m}\ C_p\ (t_o - t_i)/A_a$$

where, $q_u$ = rate of useful energy extraction from the collector, w

$A_a$ = transparent frontal area for a flat-plate collector, $m^2$

$I$ = total solar irradiation incident upon the aperture plane of collector, $w/m^2$

$(\tau\alpha)_e$ = effective transmittance-absorptance product, dimensioless

$U_L$ = solar collector heat transfer loss coefficient, $w/(°c \cdot m^2)$

$t_p$ = average temperature of the absorbing surface for a flat-plate collector, °c

$t_a$ = ambient air temperature, °c

$\dot{m}$ = mass flow rate of the transfer fluid, kg/s

$C_p$ = specific heat of the transfer fluid, $J/(°c \cdot kg)$

$t_o$ = temperature of the transfer fluid leaving the collect, °c

$t_i$ = Temperature of the transfer fluid entering the collector, °c

To assist in obtaining detailed information about the performance of the flat-plate collector and to preclude the necessity for determining the average temperature of the receiver surface, it has been convenient to introduce a parameter FR where:[13]

$$F_R = \frac{\text{actual useful energy collected by a flat-plate collector}}{\text{useful energy collected if the entire flat-plate collector surface were at the inlet fluid temperature}}$$

Introducing this factor into the original aquation results in

$$q_u/A_a = F_R\ [\ I\ (\tau\alpha)e - U_L\ (t_i - t_a)] = \dot{m}\ C_p\ (t_o - t_i)$$

If the solar collector efficiency is defined as

$$\eta = \frac{\text{actual useful energy collected}}{\text{solar energy incident upon or intercepted by the collector}} = \frac{q_u/A_a}{I}$$

then the efficiency of the flat-plate collector is given by:

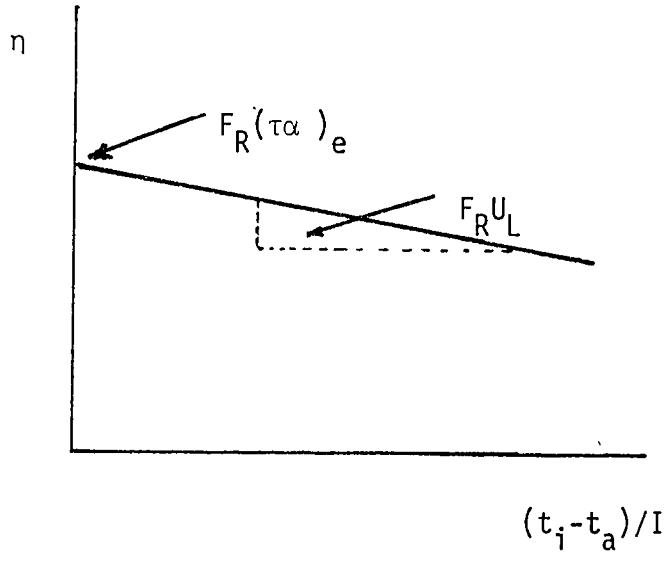
$$\eta = (A_a/A_g)\ F_R\ [(\tau\alpha)e - U_L(t_i - t_a)/I] = \dot{m}\ C_p\ (t_o - t_i)$$

where, $A_g$ = gross collector area, $m^2$

For the two collectors to be tested, $A_g = A_a = A$, hence

$$\eta = F_R\ [(\tau\alpha)e - U_L(t_i - t_a)/I] = \dot{m}\ C_p\ (t_o - t_i)/AI$$

The above equation indicates that if the efficency is plotted against $(t_i - t_a)/I$, a straight line will result where the slope is equal to $F_R$ and the y intercept is equal to $F_R(\tau\alpha)e$. In reality $U_L$ is not a constant but rather a function of the temperature of the collector and of the ambient weather conditions.



In addition, the product $(\tau\alpha)_e$ varies with the incident angle between the solar beam and the collector.

NOTE: The numerical variables required for computing thermal efficiencies of both collectors are as follows

$\dot{m} = \rho A_o V_o$

where, $\rho$ = air density (1.12 kg/$m^3$)

$A_o$ = air flow outlet area (0.012 $m^2$ for polyethylene absorber collector and 0.018 $m^2$ for cut out tin-can absorber collector)

$V_o$ = outlet air velocity, m/s

$C_p$ = 1005 J/kg·K

$A$ = 1.2 $m^2$ for both collectors

## 2.4 Results and Comments

Both collectors are tested at two air flow rates with their data tabulated in Table 2 and Table 3. Figure 7 and Figure 8 compare thermal efficiency curves of each collector with that of the ASHRAE STANDARD 93-77 curves. Figure 9 shows thermal efficiency curves for the cut out tin-can air heater (more efficient collector) tested at different flow rates. A typical steady state temperature print-out from the test is shown in Figure 10.

The results clearly show that the thermal efficiency curves for both collectors compare well with the ASHRAE STANDARD 93-77 curves. There are two distinct observations to be pointed out from these curves. Firstly, the $F_R(\tau\alpha)_e$ values (efficiency values where $t_i$ equals $t_a$) for both collectors are about the same or higher in magnitude than that of the ASHRAE curves. This can be explained by the fact that the two collectors are tested indoors under a controlled environment and under direct solar simulator insolation, whereas, the ASHRAE curves are from outdoor tests. Secondly, the values for $F_RU_L$, or the losses (slope of the curves), of both collectors are about the same or smaller than that of the ASHRAE curves. This is as predicted, since the ASHRAE collector uses a double-glazed cover and thicker insulation.

Also, the results clearly indicate that the collector using cut out tin-cans as absorber is more efficient than that using black plastic film as absorber. A more important factor is that, under the same test conditions, the temperature rise $(T_o-T_i)$ across the tin-can absorber collector is between 4°C to 5°C higher than that across plastic absorber collector. This appreciable increase in $\Delta T$ is ideally suited for solar crop drying or solar tobacco curing.

Figure 10 shows one notable observation: the aluminium foil plate temperature (7) is higher than that of the aluminium can fin temperature (8) or the steel tin-can fin temperature (9), with all three thermocouple points near to each other. This is explained by the fact that aluminium and steel fins transfer heat to the passing air better than the aluminium foil plate.

## Table 2 BLACK POLYETHYLENE FILM ABSORBER AIR HEATER TEST DATA

Constant Flow Rates Test

$\dot{m}$ = 0.015 kg/s (0.011 $m^3/s\text{-}m^2$)

| To-Ti<br>(°c) | Ti-Ta<br>(°c) | I<br>($w/m^2$) | (Ti-Ia)/I<br>(°$cm^2/w$) | η<br>(%) |
|---|---|---|---|---|
| 24.3 | -0.6 | 647 | -0.001 | 47 |
| 23.6 | 0.8 | 647 | 0.001 | 46 |
| 22.6 | 3.0 | 647 | 0.005 | 44 |
| 21.8 | 5.5 | 647 | 0.009 | 42 |
| 19.8 | 9.2 | 647 | 0.014 | 38 |
| 17.5 | 14.5 | 647 | 0.023 | 34 |
| 14.0 | 19.0 | 647 | 0.030 | 27 |
| 10.3 | 26.0 | 647 | 0.040 | 20 |
| 4.5 | 33.3 | 647 | 0.051 | 9 |

$\dot{m}$ = 0.011 kg/s (0.008 $m^3/s\text{-}m^2$)

| | | | | |
|---|---|---|---|---|
| 28.6 | 1.7 | 647 | 0.003 | 41 |
| 28.2 | 3.6 | 647 | 0.005 | 40 |
| 27.0 | 8.2 | 647 | 0.012 | 38 |
| 22.5 | 14.8 | 647 | 0.023 | 32 |
| 21.5 | 20.3 | 647 | 0.031 | 31 |
| 19.3 | 25.4 | 647 | 0.039 | 27 |
| 10.8 | 38.8 | 647 | 0.060 | 15 |
| 1.0 | 55.7 | 647 | 0.087 | 2 |

## Table 3 CUT OUT TIN-CAN ABSORBER AIR HEATER TEST DATA

### Constant Flow Rates Test

$\dot{m}$ = 0.017 kg/s (0.013 $m^3/s\text{-}m^2$)

| To-Ti<br>(°c) | Ti-Ta<br>(°c) | I<br>($w/m^2$) | (Ti-Ta)/I<br>(°$cm^2/w$) | η<br>(%) |
|---|---|---|---|---|
| 28.0 | 4.3 | 647 | 0.007 | 62 |
| 26.8 | 6.2 | 647 | 0.010 | 59 |
| 22.5 | 13.2 | 647 | 0.020 | 50 |
| 20.2 | 17.8 | 647 | 0.028 | 44 |
| 17.5 | 22.5 | 647 | 0.035 | 39 |
| 14.2 | 30.4 | 647 | 0.042 | 31 |
| 9.8 | 38.0 | 647 | 0.059 | 22 |

$\dot{m}$ = 0.01 kg/s (0.008 $m^3/s\text{-}m^2$)

| To-Ti<br>(°c) | Ti-Ta<br>(°c) | I<br>($w/m^2$) | (Ti-Ta)/I<br>(°$cm^2/w$) | η<br>(%) |
|---|---|---|---|---|
| 32.0 | 2.0 | 572 | 0.004 | 47 |
| 27.2 | 10.5 | 572 | 0.018 | 40 |
| 22.8 | 15.2 | 572 | 0.027 | 33 |
| 18.0 | 20.5 | 572 | 0.036 | 26 |
| 14.5 | 28.2 | 572 | 0.049 | 21 |
| 5.0 | 42.0 | 572 | 0.073 | 7 |

### Varying Flow Rates Test

| $\dot{m}$<br>(kg/s) | <br>($m^3/s\text{-}m^2$) | To-Ti<br>(°c) | I<br>($w/m^2$) | η<br>(%) |
|---|---|---|---|---|
| 0.036 | 0.027 | 17.6 | 647 | 82 |
| 0.035 | 0.026 | 18.0 | 647 | 81 |
| 0.034 | 0.025 | 18.3 | 647 | 80 |
| 0.033 | 0.025 | 13.6 | 647 | 80 |
| 0.027 | 0.020 | 21.0 | 647 | 74 |
| 0.022 | 0.016 | 25.2 | 647 | 71 |
| 0.017 | 0.013 | 28.0 | 647 | 62 |
| 0.015 | 0.011 | 31.5 | 647 | 61 |
| 0.006 | 0.005 | 39.8 | 647 | 32 |
| 0.004 | 0.003 | 42.0 | 647 | 22 |
| | | | | |
| 0.031 | 0.023 | 17.5 | 572 | 80 |
| 0.027 | 0.020 | 20.5 | 572 | 80 |
| 0.018 | 0.013 | 26.2 | 572 | 68 |
| 0.012 | 0.009 | 32.0 | 572 | 58 |
| 0.006 | 0.005 | 36.0 | 572 | 33 |

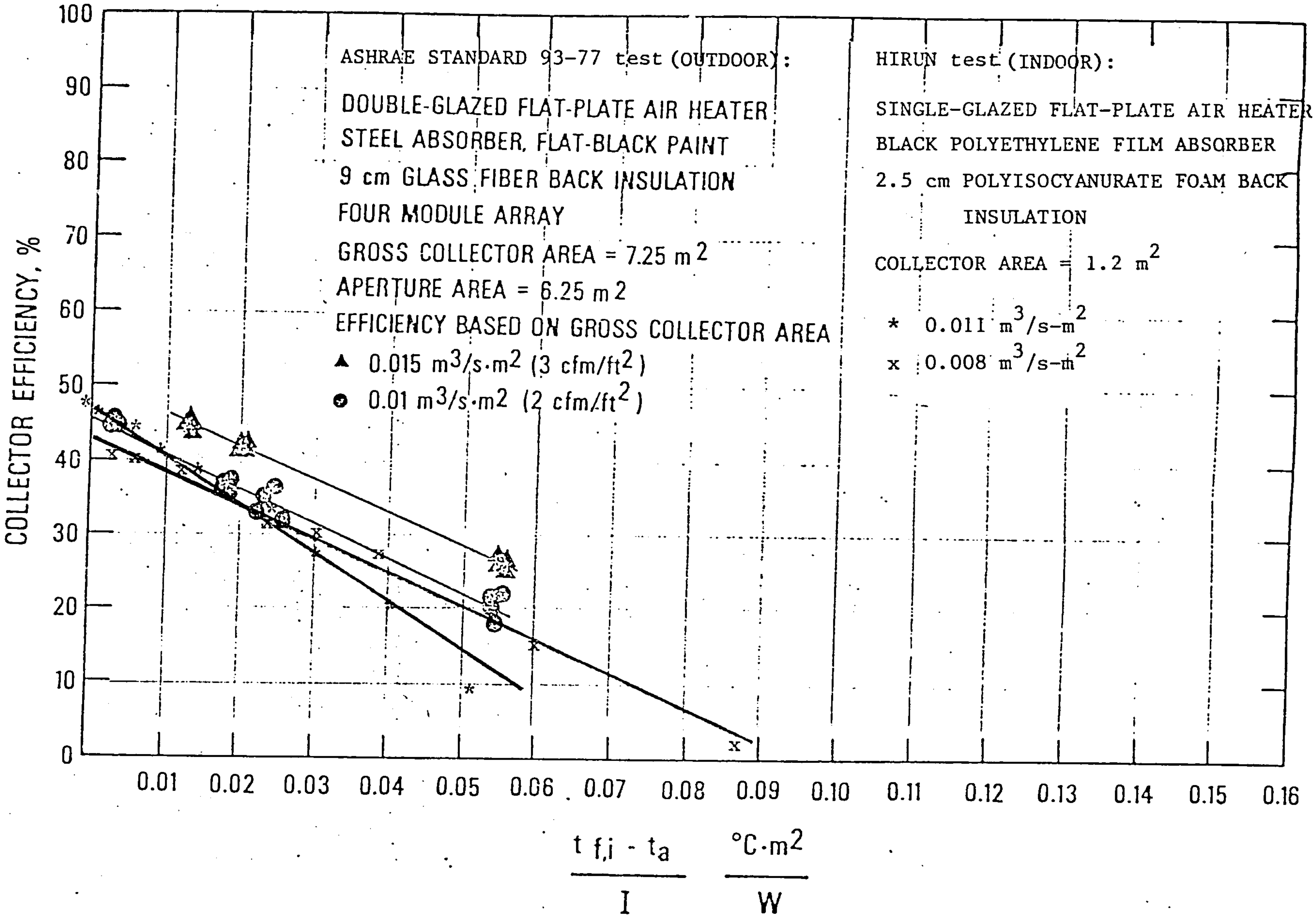


Figure 7 Thermal Efficiency Curves for Black Polyethylene Film Air Heater, a Comparison with ASHRAE STANDARD 93-77 Curves

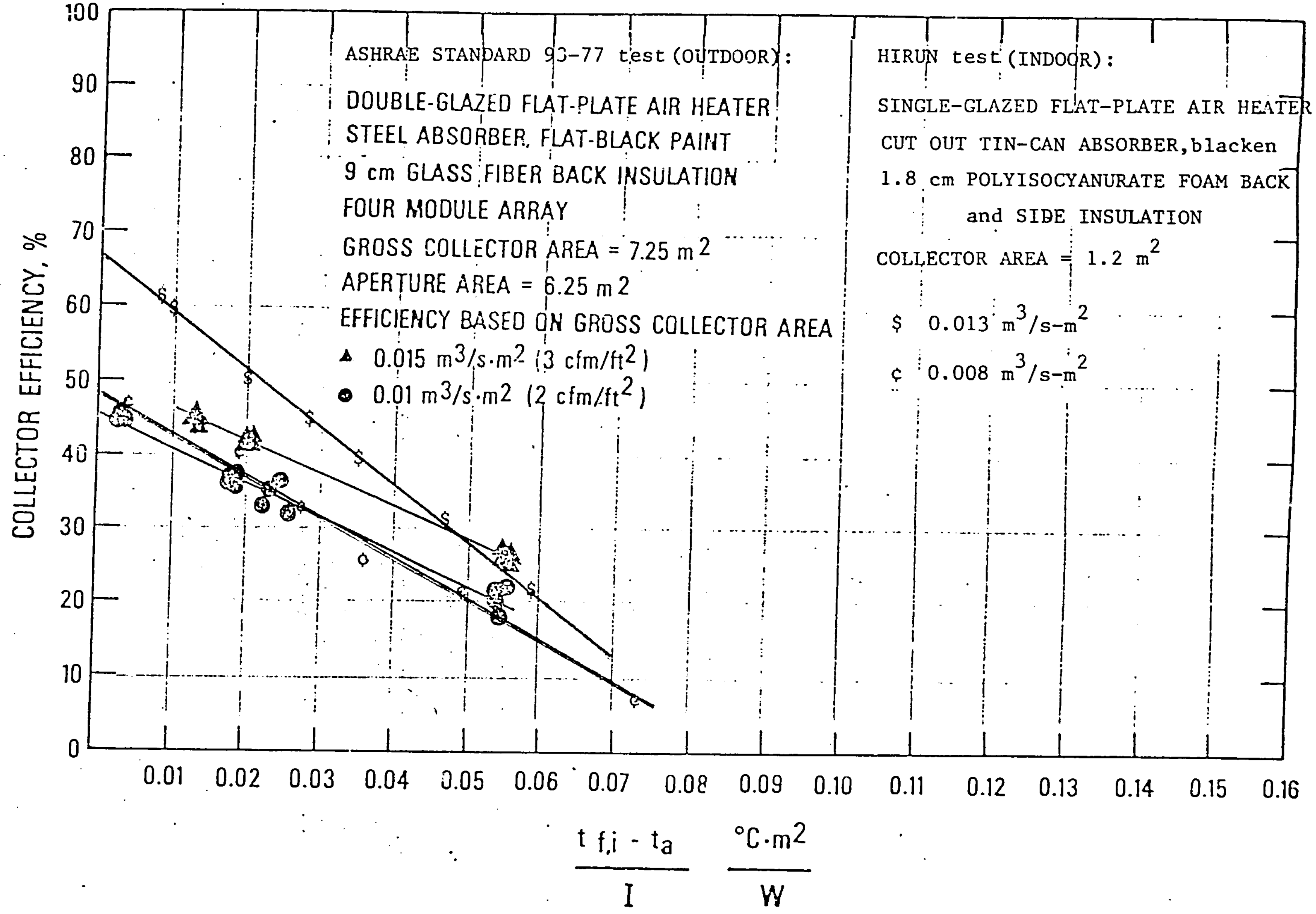


Figure 8 Thermal Efficiency Curves for Cut Out Tin-Can Air Heater, a Comparison with ASHRAE STANDARD 93-77 Curves

Figure 9 Thermal Efficiency Curves for Cut Out Tin-Can Air Heater Tested at Different Flow Rates

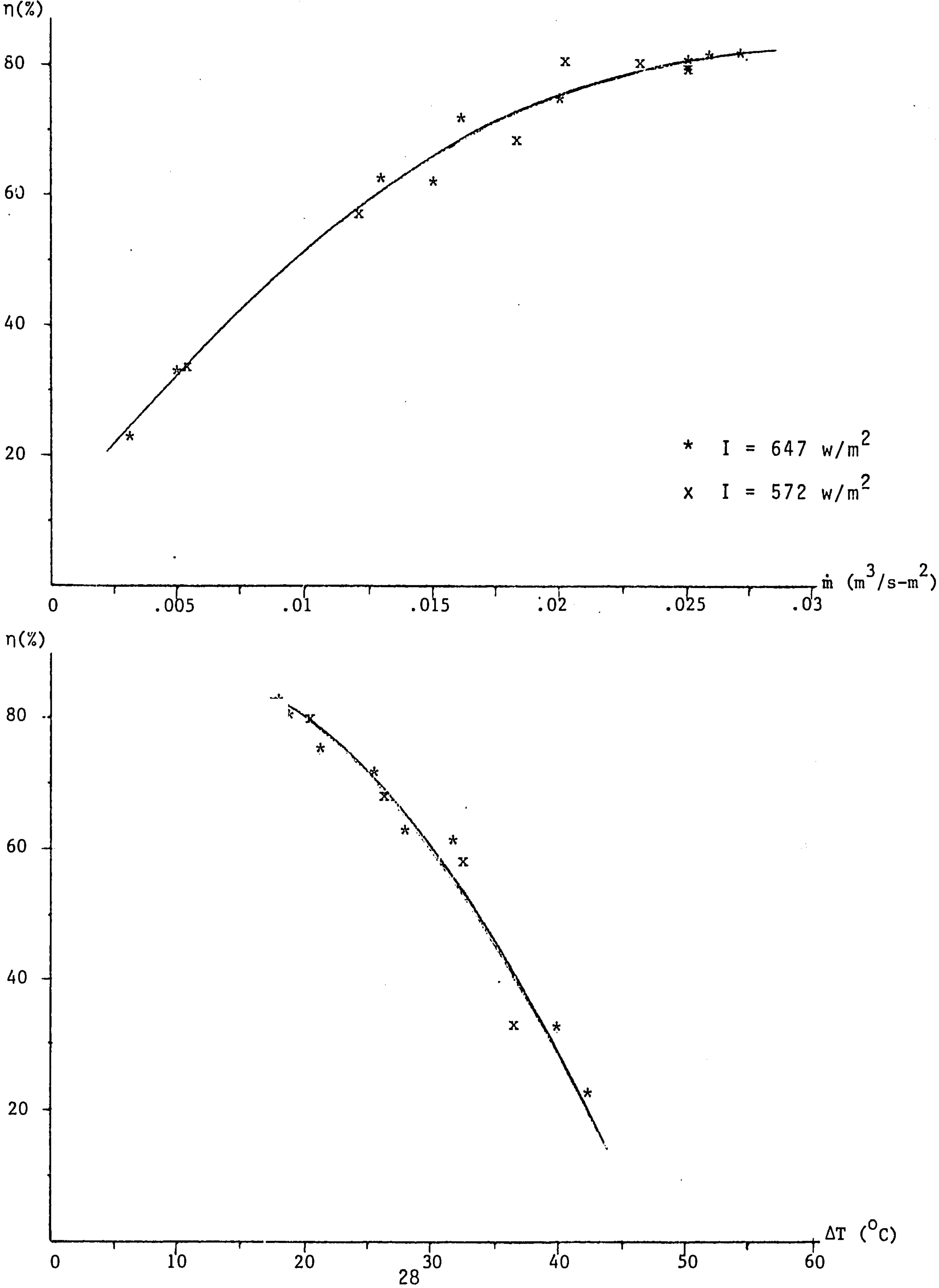
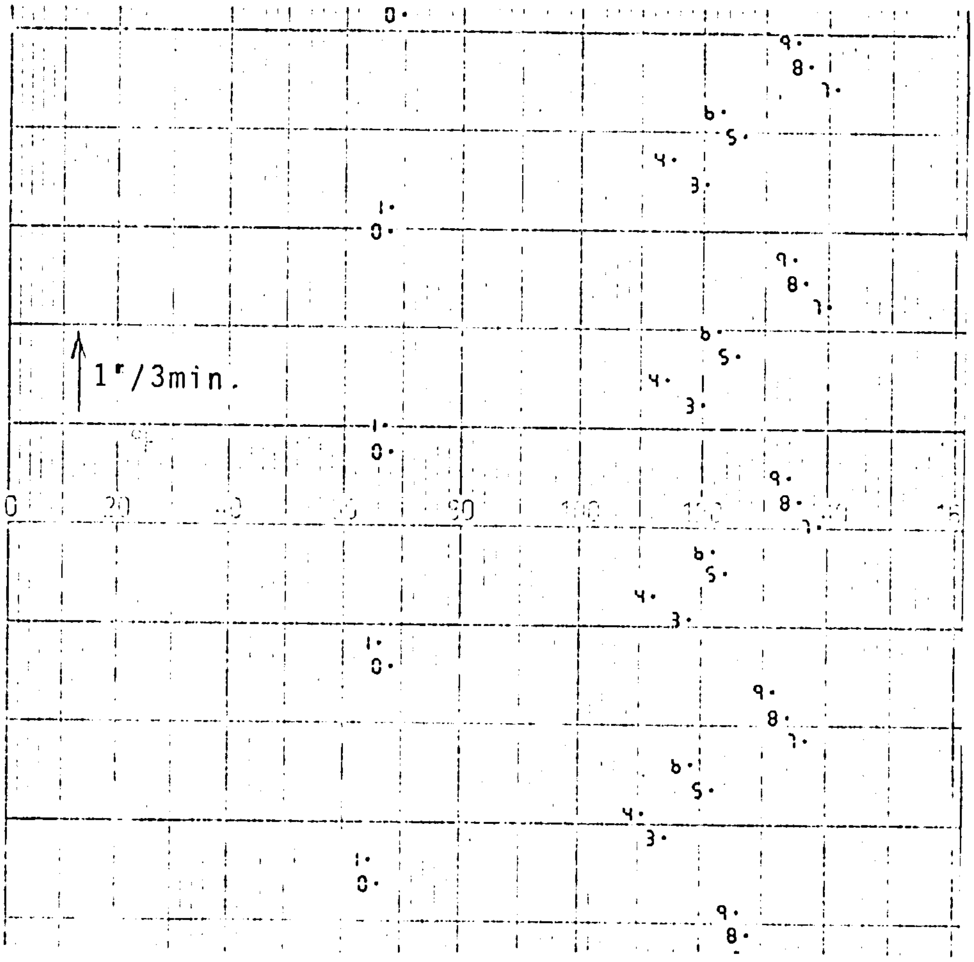


Figure 10 Typical Steady State Temperature Print-Out from the Cut Out Tin-Can Air Heater Test



Test variables:

$\dot{m} = 0.011\ m^3/s\text{-}m^2$

$I = 647\ w/m^2\ (2.33\ MJ/m^2\text{-}hr)$

$T_o - T_i = 31.5\ ^oc$

$\eta = 61\ \%$

Thermocouple points:

0=ambient air temperature

1=inlet air temperature

3=air temperature $\frac{1}{4}$L from inlet

4= " " ½L " "

5= " " $\frac{3}{4}$L " "

6=outlet air temperature

7=aluminium foil plate temperature

8=aluminium can fin temperature

9=steel tin-can fin temperature

# PART 3 Assessment and Conclusion

## 3.1 Assessment

The tests proved that the use of discarded tin-cans as absorber for a solar air heater is technically viable. The tin-cans have good heat transfer properties and they are more durable than plastic film.

It is estimated that a standard barn of 324 $m^8$ requires about 90 $m^2$ of solar collector area and a 0.25 kW fan or blower (see Appendix B). Matching this collector area with the rate of global solar radiation for Chiang Mai, Thailand, results in 5-3 MJ of solar energy contribution per kg of cured leaves, or about 15% solar energy contribution (see Appendix C).

If the above estimations hold true and by incorporating solar systems to all of the wood-fired and lignite-fired barns in Thailand, the annual energy saving will amount to over 250,000 $m^3$ of firewood equivalent or over 100,000 tonnes of lignite-equivalent.

## 3.2 Conclusion

The potential use of solar energy to implement some of the heat requirement for tobacco curing in Thailand appears logical. This investigation illustrates one approach in utilizing solar energy. The use of discarded tin-cans or other metal waste as absorbing material for the solar air heater seems practical. These materials are abundant and can cause waste disposal problems.

Further work is needed in order to confidently assess the extent of this solar energy potential. Information on tobacco curing energy requirement, air flow rate, pressure drop, etc, are non existent. This information is vitally important for assessing the solar energy potential. The investigator and his colleagues at the Department of Mechanical Engineering, Chiang Mai University, Thailand, are involved in such investigations, with financial assisstance from the government of Australia.

## Appendix A Global Solar Radiation and Climatological Data for Chiang Mai (lat.18°47'N),Thailand

Source:adapted from Exell and Saricali,"The Availability of Solar Energy in Thailand",published by AIT,Bangkok,Thailand,1975

Table A.1 Seasonal Variation of Hourly and Daily Mean Global Solar Radiation during Curing Season (MJ/$m^2$-hr)

| Apparent Time | NOV30-JAN13 | JAN14-FEB26 | FEB27-APR12 | APR13-MAY28 |
|---|---|---|---|---|
| 7-8 | 0.46 | 0.67 | 0.75 | 1.05 |
| 8-9 | 1.13 | 1.34 | 1.42 | 1.76 |
| 9-10 | 1.76 | 1.97 | 2.01 | 2.35 |
| 10-11 | 2.22 | 2.47 | 2.56 | 2.72 |
| 11-12 | 2.43 | 2.72 | 2.81 | 3.02 |
| 12-13 | 2.39 | 2.72 | 2.81 | 2.89 |
| 13-14 | 2.18 | 2.43 | 2.51 | 2.51 |
| 14-15 | 1.72 | 1.89 | 1.93 | 1.93 |
| 15-16 | 1.13 | 1.26 | 1.30 | 1.38 |
| 16-17 | 0.54 | 0.59 | 0.67 | 0.80 |
| Totals(MJ/$m^2$day) | 15.96 | 18.06 | 18.77 | 20.41 |

Table A.2 Climatological Data for Chiang Mai during Curing Season

| Month | NOV | DEC | JAN | FEB | MAR | APR |
|---|---|---|---|---|---|---|
| Mean Temp.(°C) | | | | | | |
| Daily Max. | 29.8 | 28.5 | 29.0 | 32.1 | 34.9 | 36.2 |
| Daily Min. | 18.6 | 14.7 | 13.0 | 13.8 | 17.2 | 21.1 |
| Mean R.H.(%) | 80.0 | 77.0 | 74.0 | 65.0 | 58.0 | 60.0 |
| Mean Wind Speed (km/hr) | 3.3 | 3.2 | 3.5 | 4.4 | 5.4 | 6.7 |

## Appendix B Fan and Collector Sizing

known data:

- barn volume = 6 x 6 x 9 = 324 $m^3$
- barn density = 1500/324 = 4.63 kg of fresh tobacco/$m^3$

assume data:

- air flow rate inside the barn = 0.15 $m^3$/min-$m^3$ volume
- total air flow rate = 0.15 x 324 = 48.6 $m^3$/min = 0.81 $m^3$/s
- pressure drop = 1.5 m air per m depth (this includes fan entrance and diffuser losses, this value is ½ that of grain
- total pressure drop = 1.5 x 9 = 13.5 m air
- fan efficiency = 65 %
- mean solar radiation = 1.8 MJ/$m^2$-hr = 500 W/$m^2$ = 160 $\frac{BTU}{ft^2\text{-}h}$ (derived from Table A.1)

Fan Sizing:

$$\text{Fan Power Rating}, P = \frac{\rho g Q H}{\eta} = \frac{1.12(kg/m^3) \times 9.81(m/s^2) \times 0.81(m^3/s) \times 13.5(m)}{0.65}$$

$$= 185\ W \quad (¼\ HP)$$

Note: for overloading and safety reasons use P = 250 W

Collector Sizing:

$$q_{useful} = \dot{m} C_p (t_o - t_i)$$

where,

$q_{useful}$ = efficiency x insolation rate

$\dot{m}$ = mass air flow rate per $m^2$ of collector area

$C_p$ = 1005 J/kg-°C (specific heat of air)

$(t_o - t_i)$ = temperature rise across the collector

Note: $t_o - t_i$ = 25°C is sufficient for first two stages of curing

From Figure 9, at $t_o - t_i$ = 25°C, Eff. = 70 % (use 50% for outdoors)

thus, $$\dot{m} = \frac{500(J/s\text{-}m^2) \times 0.5}{1005(J/kg\text{-}^\circ C \times 25^\circ C} = 0.01\ kg/s\text{-}m^2$$

$$= \frac{0.01(kg/s\text{-}m^2)}{1.12(kg/m^3)} = 0.009\ m^3/s\text{-}m^2$$

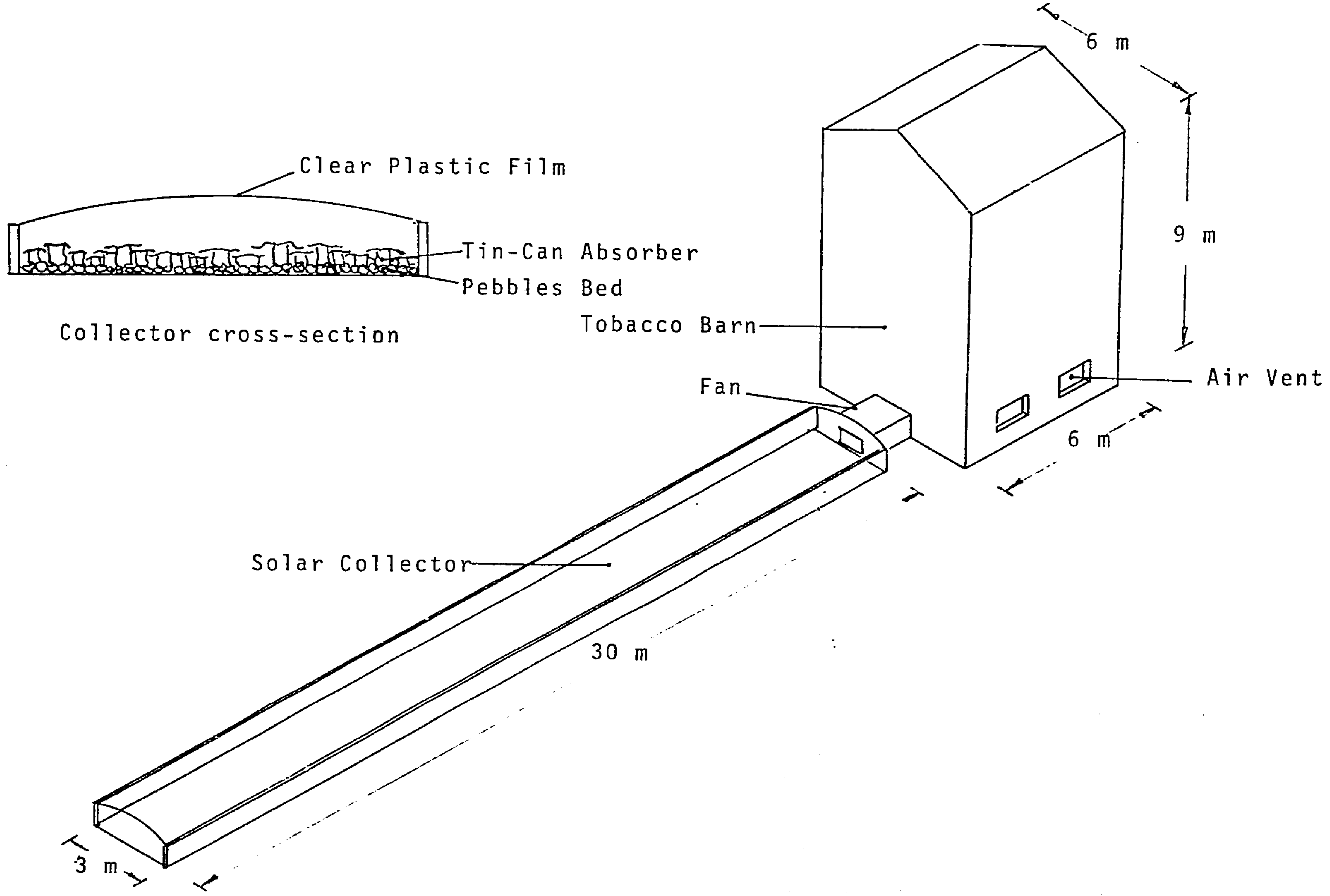
Note: from Figure 9, at $\dot{m}$ = 0.009 $m^3/s\text{-}m^2$, Eff. = 50 %

For total air flow = 0.81 $m^3$/s,

$$\text{Collector Area} = \frac{0.81(m^3/s)}{0.009(m^3/s\text{-}m^2)} = 90\ m^2$$

(See Figure B for the projected view of the solar assisted tobacco curing system)

Figure B Projected View of the Solar Assisted Tobacco Curing System



## Appendix C Solar Energy Contribution

Estimation of solar energy contribution per kg of cured leaves,

collector area per barn = 90 $m^2$ (see Appendix B)
collector efficiency = 30 % (assumption)
mean daily solar radiation = 18.3 $MJ/m^2$-day (see Table A.1)
maximum curing period per batch = 4 days (see Table 1)
kg of cured leaves per batch per barn = $\frac{1500}{4}$ = 375

Thus,

$$\text{solar energy contribution} = \frac{18.3(MJ/m^2\text{-day})\times 4(\text{days})\times 90(m^2)\times 0.3}{375(kg)}$$

$$= 5.3 \text{ MJ/kg of cured leaves}$$

Note: Cundiff, 1978, reported that 20 MJ heat energy input is needed per kg of cured leaves.* This will result in about 25 % solar energy contribution.

Source*: Cundiff, J.S. (1978). Fan cycling and energy consumption in bulk tobacco curing. Univ. of Ga., College of Agric. Res. Rep. No. 288

## REFERENCES

1 National Bank of Thailand (1983). Monthly Economics Report Vol. 1, January issue.

2 Boon-loog, Hirun, Theerakulpisut, (1983). Energy Conservation in Tobacco Leaves Curing in Northern Thailand. A financial assistance proposal report to the Government of Australia.

3 Johnson, W.H. (1980). Energy Conservation in tobacco curing with special reference to the use of solar energy. North Carolina State University, Raleigh.

4 Cundiff, J.S. (1980). A renewable resources system for curing tobacco. University of Georgia, Coastal Plain Experiment Station, Agricultural Engineering Department, Tifton.

5 Necesito, A.C. (1978). The energy requirement of curing Virginia tobacco in the Philippines and the potential use of solar energy. Proceedings of the Solar Drying Workshop, Manila, Philippines.

6 See [2]

7 Bamrungwong, S. (1980). A study and design of solar heating systems for tobacco curing. M. Eng. Thesis, Graduate School, Chulalongkorn University, Thailand.

8 See [2]

9 See [4]

10 Watkins, R.W. (1978). Mechanization ' . Tobacco Report 48-51.

11 ASHRAE, (1977). Methods of testing to determine the Thermal Performance of Solar Collectors. ASHRAE STANDARD 93-77, New York.

12 Laketek, L.E. (1983). Solar and Ancillary Measurement Equipments, Training in Alternative Energy Technologies (TAET). USAID and University of Florida, Gainesville.

13 See [11]