PARAMETERS OF SOLAR COLLECTORS TESTED WITH KMITT SOLAR SIMULATOR

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Abstract

This paper reports on the experimental performances of flat plate solar collectors tested with a solar simulator under steady-state conditions in terms of collector efficiency, η, and ratio of temperature difference and solar radiation, $(T_{fi} - T_e)/I_T$. T_e was the effective heat - sink temperature of the tested collector and could be evaluated from temperatures of collector's cover, ambient and light source panel (or infrared filter). Techniques for converting values of the collector's parameters, $F_R U_{Le}$ and $F_R(\tau\alpha)_e$, obtained from the indoor tests to match outdoor results were demonstrated. The adjusted results agreed well with those of the outdoor data in the case of a collector having a flat glass cover. For a collector having a convex plastic cover, the estimated optical efficiency was lower than that of the outdoor result.

INTRODUCTION

The steady-state approach to test solar collector performance has been widely used. However, it is realised that the procedure is time-consuming due to variation of solar radiation flux. Clear sky periods for experiments are rarely available, especially in tropical regions. Thus to obtain a full set of data sometimes several weeks for testing are needed. A solar simulator is a light source from which the spectral output is close to real solar radiation and the radiation flux can be adjusted at different levels. Solar equipment can be tested indoor continuously and a whole set of experimental data can be achieved in a much shorter period when compared with that of outdoor testing.

A solar simulator has been constructed at King Mongkut's Institute of Technology Thonburi (KMITT) in Thailand and operated since 1988. The system as shown in Fig. 1 consists of an air-cooled lamp array of 2 x 2 m^2, an infrared filter and a collector test rig. The lamp array mounts 115 OSRAM HQT-R 250 W/DL lamps which can generate radiation over 810 W/m^2 on tested collector plane. The infrared filter consists of double 1.95 x 1.95 m^2 clear glass panes each of 6 mm thickness and 3 mm apart. Clean water is fed through the gap to control the filter temperature to be close to that of the surrounding ambient. Thus the effect of longwave radiation emitted from the filter to the tested collectors is alleviated.

Under steady state conditions, parameters of flat plate collectors are always plotted in terms of efficiecy, η, with ratio of temperature difference over incidence radiation, $(T_{fi} - T_a)/I_T$. Intercept on the axis and slope of the curve are $F_R(\tau\alpha)$ and F_RU_L which show optical performance and thermal loss of each collector, respectively.

Some flat plate collectors were tested with the KMITT solar simulator [1]. It was found that the radiation obtained was highly diffused, and there is an effect of thermal radiation exchange between the IR filter and the tested collector, thus some deviations from outdoor test results were obtained. These observations were also found in some solar siumlator laboratories which used a set of lamps as a light source [2,3].

The experiment described in this paper introduced an effective temperature, T_e, which is the heat-sink temperature of the tested collector in the KMITT solar simulator laboratoroy. This value is averaged from the collector's surrounding temperature and the temperature of the IR filter. A technique to convert the indoor test results to be comparable with those of the outdoor test was also demonstrated.

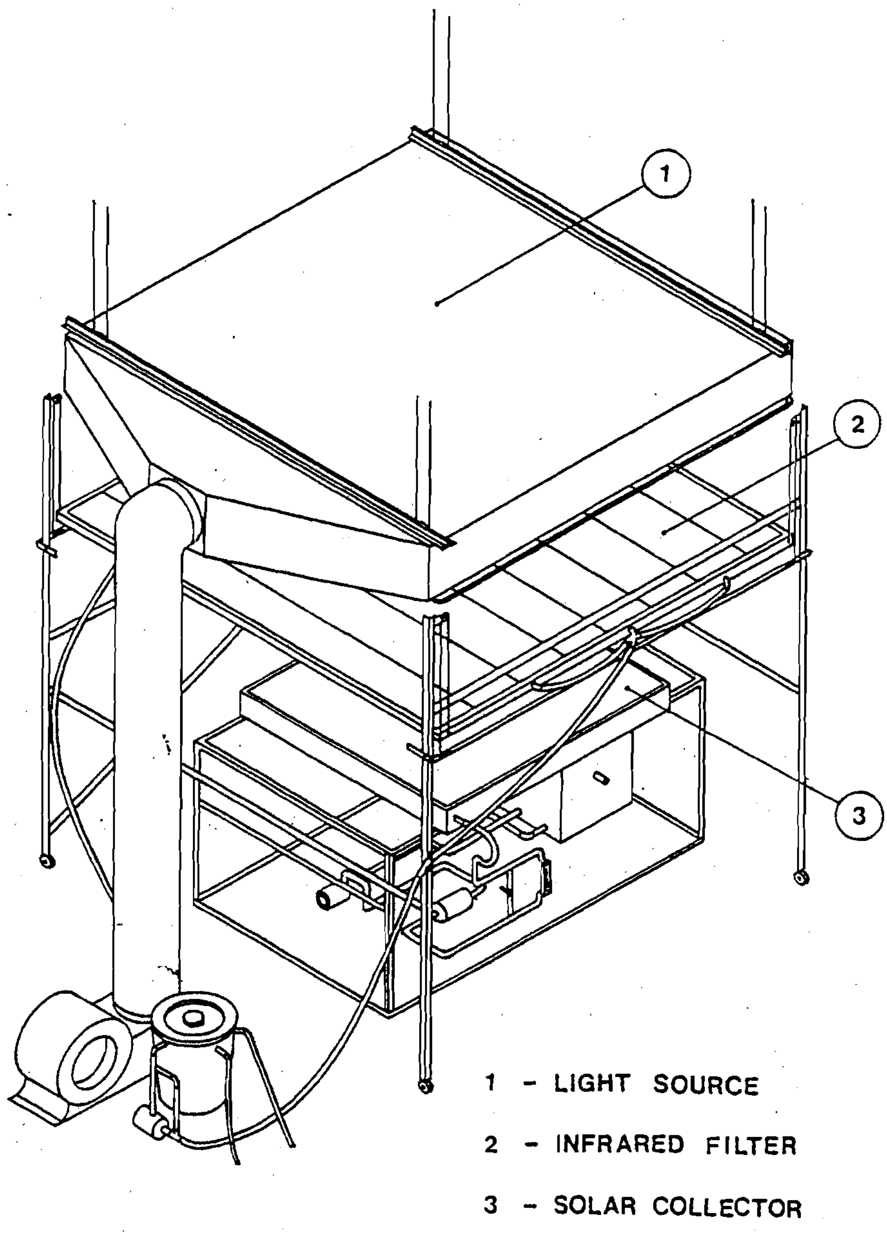

Fig.1 The KMITT Solar Simulator.

SOLAR COLECTOR EFFECTIVE HEAT-SINK TEMPERATURE

At the KMITT solar simulator laboratory, an IR fillter is placed between the lamp array and the tested collector as described in the previous section. Therefore, there are thermal losses from the collector to the surrounding ambient by convection and radiation, including thermal radiation loss to the IR filter. A diagram of thermal losses under steady stale condition of a tested collector is shown in Fig. 2.

Rate of thermal losses from upper part of collector per unit area

$$= h_{c,p-c}(T_p-T_c) + h_{r,p-c}(T_p - T_c)$$

$$= h_{c,c-a}(T_c-T_a) + h_{r,c-a}(T_c-T_a) + h_{r,c-filt}(T_c-T_{filt}), \quad (1)$$

or

$$T_c = (AT_p + BT_a + CT_{filt})/(A + B + C), \quad (2)$$

where

$A = h_{c,p-c} + h_{r,p-c}$,

$B = h_{c,p-c} + h_{r,c-a}$,

$C = h_{r,c-filt}$,

$h_{c,c-a} = 2.8 + 3V$; V is wind speed in m/s.

Rate of thermal losses from back of the collector per unit area

$$= U_{b,p-a}(T_p-T_a). \quad (3)$$

From eqns. (1) and (2), rate of total heat losses per unit area, q_{loss} can be written as :

$$q_{loss} = (h_{c,p-c} + h_{r,p-c})(T_p-T_c) + U_{b,p-a}(T_p-T_a). \quad (4)$$

Substitute T_c from eqn. (2) into eqn. (4),

$$q_{loss} = [AB/(A+B+C) + U_{b,p-a}](T_p - T_a) + [AC/(A+B+C)](T_p - T_{filt}), \quad (5)$$

Therefore, the rate of useful heat-gain per unit area, q_u, can be written as :

$$q_u = (\tau\alpha)_e I_T - U_{La}(T_p - T_a) - U_{Lf}(T_p - T_{filt}), \quad (6)$$

where

$$U_{La} = AB/(A+B+C) + U_{b,p-a},$$

$$U_f = AC/(A+B+C).$$

then

$$q_u = F_R(\tau\alpha)_e I_T - F_R U_{La}(T_{fi} - T_a) - F_R U_{Lf}(T_{fi} - T_{filt}). \quad (7)$$

F_R is the ratio of actual useful heat to useful heat when the temperature of the whole absorber was at the inlet fluid temperature.

Collector efficiency, η, then can be written as :

$$\eta = q_u/I_T = F_R(\tau\alpha)_e - F_R(T_{fi} - T_a)/I_T - F_R U_{Lf}(T_{fi} - T_{filt})/I_T. \quad (8)$$

It can be seen that, the efficiency now depends not only on T_{fi} and T_a but also T_{filt}. Therefore, a relation of η with $(T_{fi} - T_a)/I$ as the outdoor result seems to be imperfect. Eqn. (8) may then be developed in a form as :

$$\eta = F_R(\tau\alpha)_e - F_R U_{Le}(T_{fi} - T_e)/I_T \quad (9)$$

T_e is called effective heat-sink temperature of the collector, and U_{Le} is effective heat loss from the collector to ambient at the temperature, T_e.

Comparing eqns. (8) and (9), we get :

$$F_RU_{Le} = F_RU_{La} + F_RU_{Lf}, \tag{10}$$

and

$$T_e = (F_RU_{La}/F_RU_{La})T_a + (F_RU_{Lf}/F_RU_{Le})T_{filt} \tag{11}$$

When the ambient temperature T_a and the infrared filter temperature T_{filt} are measured, the effective temperature T_e could be evaluated from eqn.(11). In case of the indoor test, from eqn (9), experimental curve could be drawn in a form of η v.s. $(T_{fi}-T_e)/I_T$ diagram instead of η v.s.$(T_{fi}- T_a)/I_T$. The intercept on the η- axis is $F_R(\tau\alpha)_e$ and its slope is - F_RU_{le}.

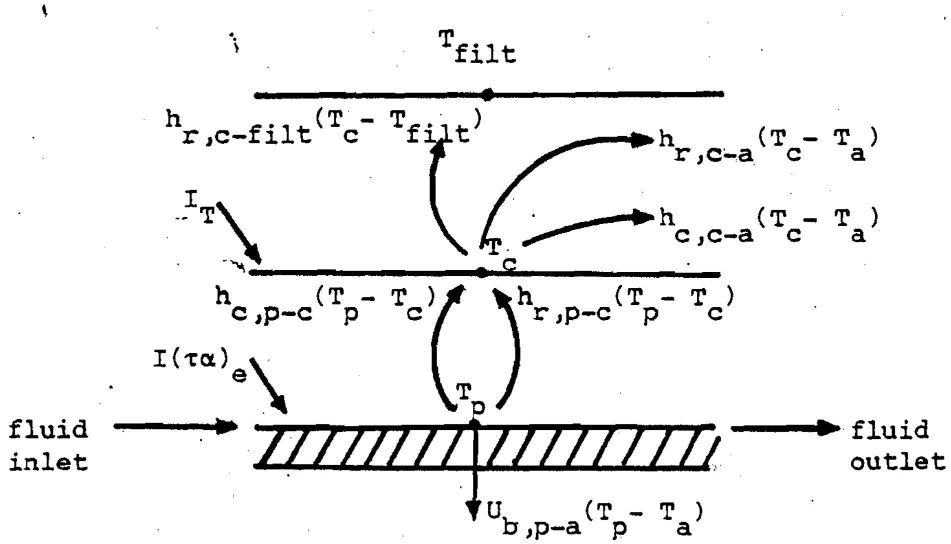

Fig. 2 Directions of energy flow of indoor tested collector.

STEPS FOR CONVERTING INDOOR PARAMETERS TO OUTDOOR RESULTS

Performance parameters of collectors tested with the KMITT solar simulator might be different from those of the outdoor tests. There are two main factors as follows :

a. The light source consists of a set of lamps and thus most of the energy emitted is diffuse radiation.

b. There is thermal radiation exchange between the collector tested and the infrared filter. In case of the outdoor tests, radiation loss from the collector is transferred to the surrounding ambient.

With these problems, the two parameters, $F_R(\tau\alpha)_e$ and F_RU_{Le} will deviate from those obtained from the outdoor experiments. Thus it is necessary to develop a technique to convert the indoor results to match those of outdoor tests. The essential features of the method can be summarized as follows :

a. The optical parameter

Outdoor tests are usually done in the mid-day hours, on clear days, with the beam solar radiation nearly normal to the collector. The transmittance-absorptance term for the test conditions is approximately the normal incidence value, and is written as $(\tau\alpha)_n$. In the case of indoor conditions, since the radiation obtained from the light source panel is highly diffused, the effective angle of incidence at the same transmittance as that of a beam radiation be evaluated by [4]

$$\theta_e = 59.68 - 0.1388\ B + 0.001497\ B^2 \qquad (12)$$

B is a slope of the collector. With the KMITT indoor test, the tested collector is oriented parallel to the IR filter and the light source panel. Thus $B = 0$ and θ_e becomes $59.68^\circ \simeq 60^\circ$.

With the value of $\theta_e = 60^\circ$, a relation between the indoor transmittance-absorptance product $(\tau\alpha)_e$ and that of the normal incidence as the outdoor condition could be estimated from Fig. 3 [5]. In the case of single glass cover, we get $(\tau\alpha)_e/(\tau\alpha)_n = 0.83$ or $F_R(\tau\alpha)_e/F_R(\tau\alpha)_n = 0.83$.

With this technique, as the $F_R(\tau\alpha)_e$ from the indoor experiment is estimated, the result can be converted to match that of the outdoors by :

$$F_R(\tau\alpha)_n = [(\tau\alpha)_n/(\tau\alpha)_e]F_R(\tau\alpha)_e. \tag{13}$$

In case of the single glass cover,

$$F_R(\tau\alpha)_n = (1/0.83)F_R(\tau\alpha)_e.$$

b. The loss parameter

The loss parameter F_RU_{Le} which is obtained from the indoor result can be converted to match that of the outdoors by :

$$\begin{aligned} F_RU_L/F_RU_{Le} &= U_L/U_{Le} \\ &= [q_{loss}/(T_p - T_a)]/[q^*_{loss}/(T_p - T_e)] \\ &= (q_{loss}/q^*_{loss})[(T_p - T_e)/(T_p - T_a)]. \end{aligned} \tag{14}$$

F_RU_L is the loss parameter obtained as if the collector is tested under outdoor conditions at the temperature T_a. The terms q_{loss} and q^*_{loss} are the rates of heat loss from the collector under outdoor and indoor conditions, respectively, which can be evaluated as follows :

$$q_{loss} = \varepsilon_c\sigma[(T_c + 273)^4 - (T_a + 273)^4] + h_{c,c-a}(T_c - T_a) + q_{back} \tag{15}$$

and

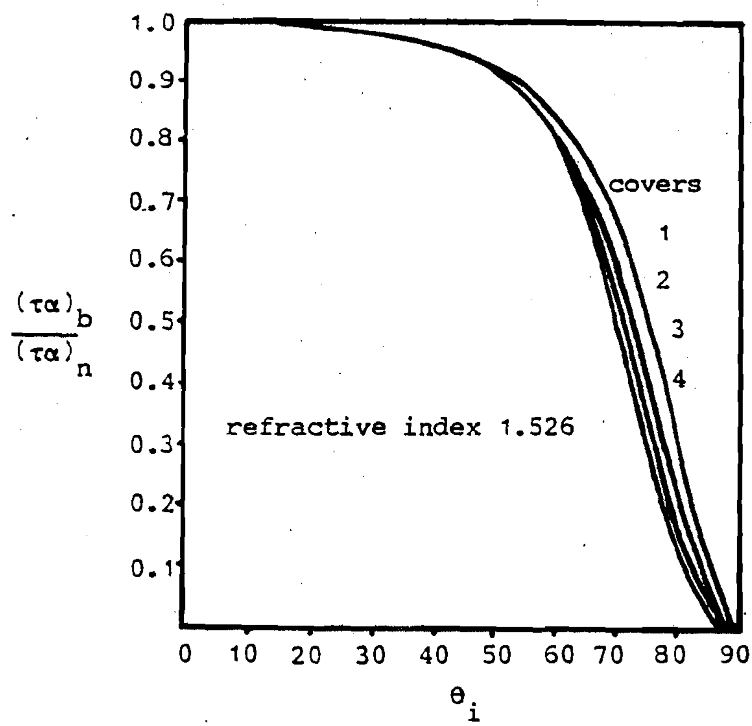

Fig. 3 Typical $(\tau\alpha)_b/(\tau\alpha)_n$ curves for glass covers.

$$q^*_{loss} = (E_{bc} - J_c)/[(1- {}_c)/({}_c] + h_{c,c-a}(T_c - T_a) + q^*_{back} \qquad (16)$$

E_{bc} and J_c are radiation emissive power of blackbody and radiosity at the cover temperature. With the values of T_c, T_{filt}, T_a and T_p, q^*_{loss} and $F_R U_L$ can be evaluated.

EXAMPLES

Two types of flat plate collectors having water as a working fluid were tested. The first one consisted of black copper absorber of $1.318 \times 1.09\ m^2$ and a single glass cover. The second one had a block rubber absorber with a convex plastic cover. Its aperture area was $1.015 \times 0.615\ m^2$.

The outdoor data of both collectors at wind speed of 3 m/s were as follows :

Black copper collector:

$$\eta = 63.1 - 892.5\ (T_{fi} - T_a)/I_T,\ (\%)$$
$$F_R(\tau\alpha)_n = 0.63,\ F_R U_L = 8.93\ W/m^2K \text{ at } T_a = 34^oC$$

Black rubber collector :

$$\eta = 68.7 - 1364\ (T_{fi} - T_a)/I_T,\ (\%)$$
$$F_R(\tau\alpha)_n = 0.69,\ F_R U_L = 13.64\ W/m^2K \text{ at } T_a = 30^oC.$$

Fig. 4 shows the performance curves of the collectors undrr outdoor conditions.

From the indoor experiments :

a) The indoor conditions for the black copper collector were as follows :

$$T_{filt} = 39.5^{o}C,\ T_a = 35^{o}C, \text{ wind speeed } 3 \text{ m/s},$$

Thermal emittances of the infrared filter and the cover of the collector = 0.85,
Configuration factor of the collector - IR fitter = 0.89

The effective temperature was calculated, with the details given in appendix a, to be 36.3°C while the average cover temperature during the experiment was 43.9°C. The relation between the collector efficiency and the value $(T_{fi} - T_e)/I_T$ was :

$$\eta = 51.04 - 788.4\,(T_{fi} - T_e)/I_T, \quad (\%)$$
$$F_R(\tau\alpha)_e = 0.51 \text{ and } F_RU_{Le} = 7.88 \text{ W/m}^2\text{K}.$$

From appendix b, the rate of heat loss from the collector under both indoor and outdoor conditions were estimated to be :

$$q^*_{loss} = 160.22 \text{ W/m}^2 \text{ and } q_{loss} = 199.62 \text{ W/m}^2.$$

The absorber temperature, T_p, normally, was higher than that of the working fluid by about 3-5°C. While the average temperature of the water during the experiment was 58.7°C, by assuming 5°C higher, the average absorber temperature, T_p, then became 63.7°C. With the values of q_{loss}, q^*_{loss} and T_p, the heat loss parameter of the indoor experiment, F_RU_{Le}, could be converted to be that of the outdoor result at the ambient temperature of 34°C by :

$$F_RU_L/F_RU_{Le} = (q_{loss}/q^*_{loss})[(T_p - T_e)/(T_p - T_a)]$$

$$= 199.62/160.22)[(63.7-36.3)/(63.7-34)]$$
$$= 1.149$$

For $F_R U_{Le} = 7.88\ W/m^2$ then $F_R U_L = 1.149 \times 7.88 = 9.06\ W/m^2\ K$.

The optical parameter obtained from the indoor experiment could be converted to match that of the outdoor by using eqn (13), thus :

$$F_R(\tau\alpha)_n = F_R(\tau\alpha)_e/0.83$$
$$= 0.51/0.83 = 0.614$$

Therefore, the evaluated outdoor performance of the copper collector could be :

$$\eta = 61.4 - 906\ (T_{fi} - T_a)/I_T \quad (\%),$$

which agreed well with the actual outdoor data. Comparisons of the indoor test results with the outdoor data are shown in Fig. 5.

b) The indoor conditions for the rubber collector were as follows :

$$T_{filt} = 39.4^{\circ}C,\ T_a = 34^{\circ}C,\ \text{wind speed } 3\ m/s$$

The collector equation at indoor conditions was :

$$\eta = 50.4 - 1045(T_{fi} - T_e)/(I_T, \quad (\%)$$
$$F_R(\tau\alpha)_e = 0.504 \text{ and } F_R U_{Le} = 10.45\ W/m^2K$$

With the method as described before, the performance curve could be adjusted to match that of the outdoor conditions as :

$$\eta = 60.7 - 1429\ (T_{fi} - T_a)\ (\%).$$

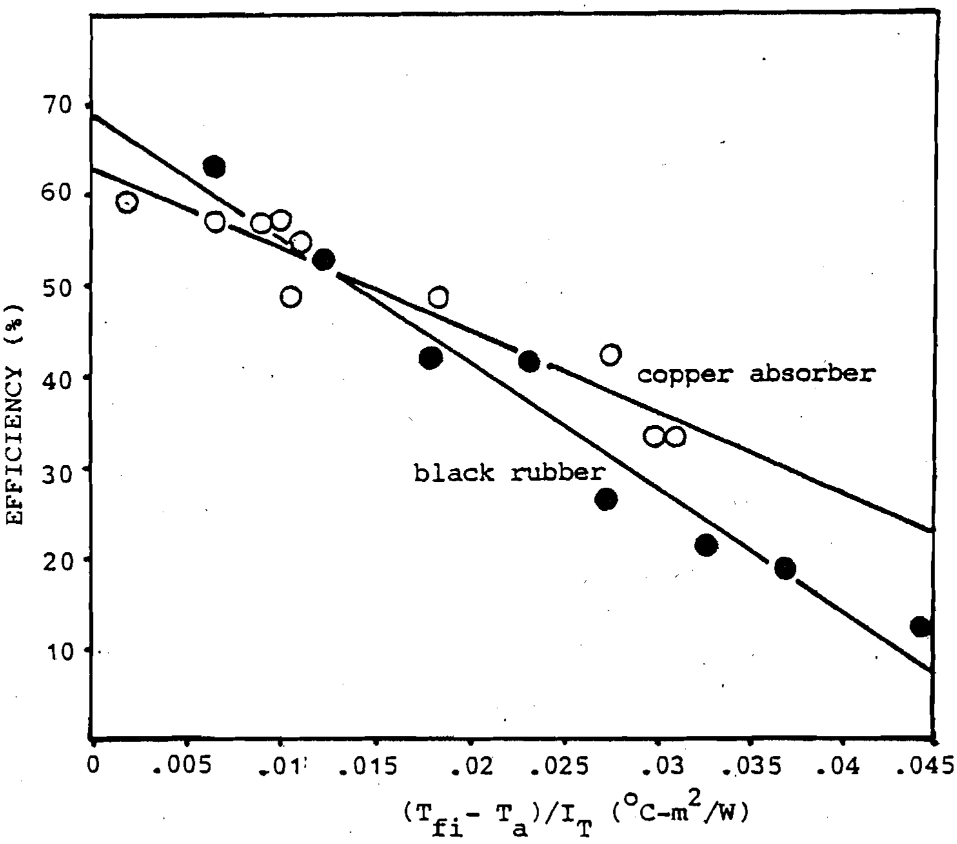

Fig. 4 Outdoor collector performance curves.

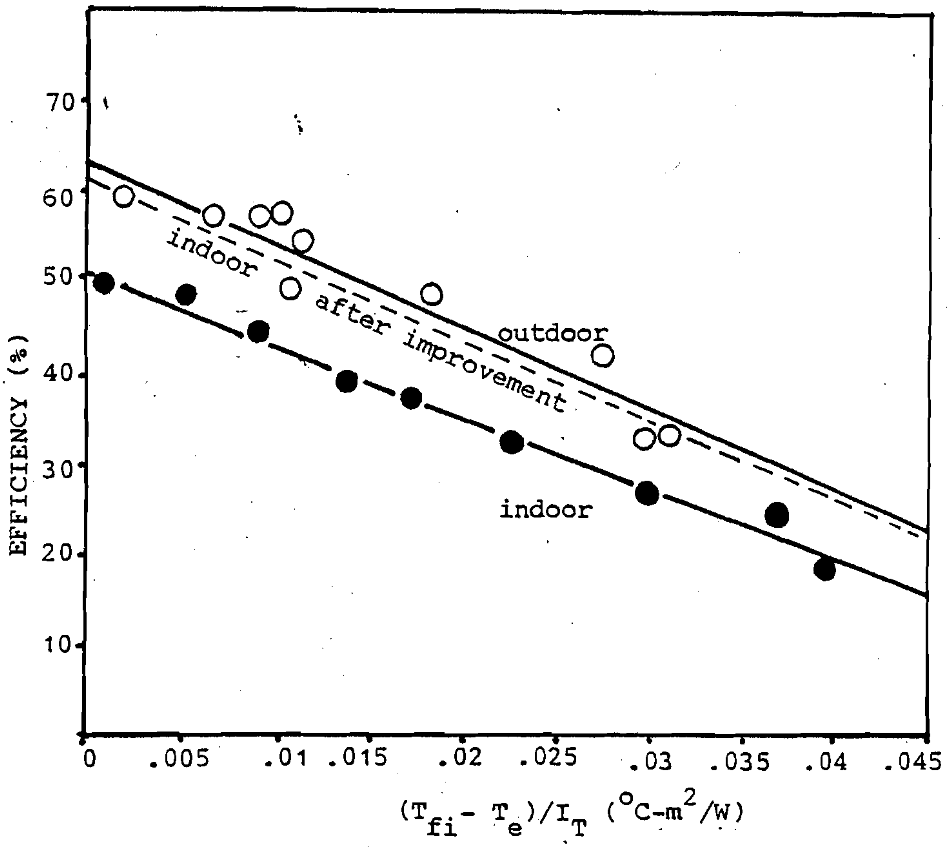

Fig. 5 Comparison of indoor and outdoor results of the copper absorber collector.

Compared with the actual data, the heat loss term agreed well with that of the actual outdoor experiment, whereas the optical term was lower since the plastic cover was convex and not flattened. The results are shown in Fig. 6.

CONCLUSIONS

The following conclusions can be drawn be drawn from the present investigations :

a) For indoor collector testing, the parameter curve of the tested collectors are proposed to be expressed in a form of η and $(T_{f1} - T_e)/I_T$ where T_e is the effective heat-sink temperature of the collectors.

b) This paper describes techniques to convert the collector parameters obtained from the indoor results to match those of outdoor conditions. It was found that the results in the case of the collecter having flat cover agreed well with those of the actual outdoor experiments.

NOMENCLATURE

A_1	Collector area, m^2
A_2	Infrared filter area, m^2
E_{bc}, E_{b1}	Emissive power of blackboby at the collector's cover temperature, W/m^2
E_{b2}	Emissive power of blackboby at the infrared filter temperature, W/m^2
E_{b3}	Emissive power of blackboby at the surrounding temperature, W/m^2
F_{1-2}	Shape factor 1-2
F_{1-3}	Shape factor 1-3

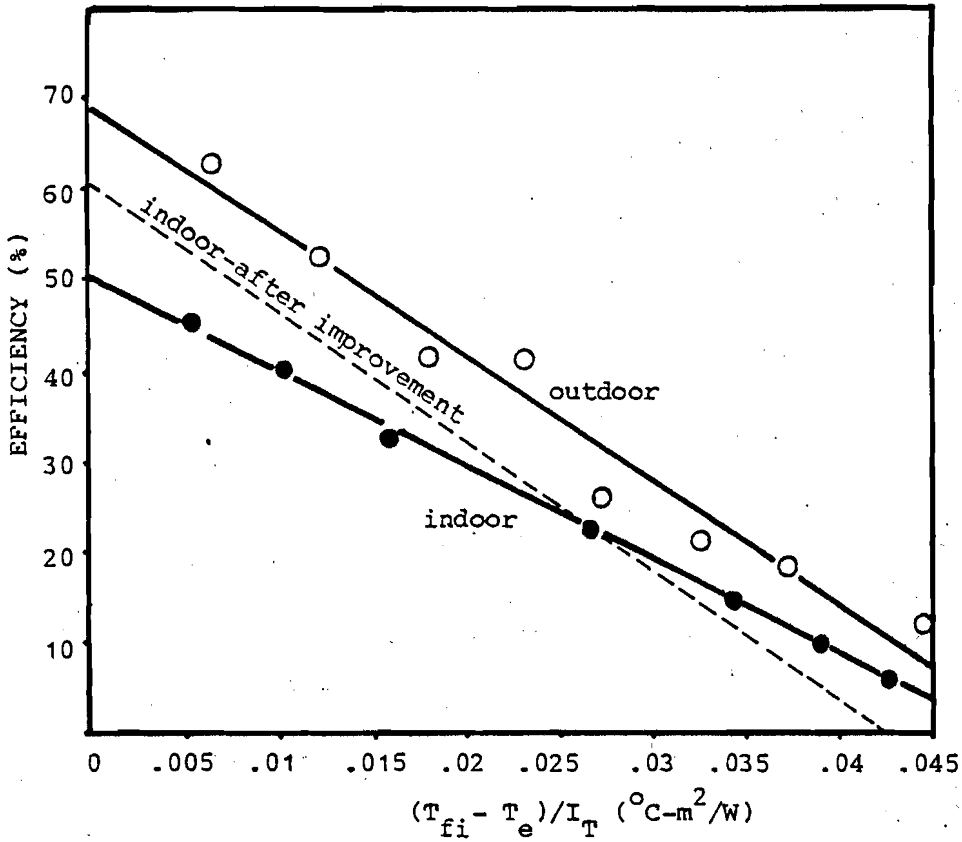

Fig. 6 Comparison of indoor and outdoor results of the rubber absorber collector.

F_R	Heat removal factor
h_c	Convective heat transfer coefficient, W/m^2K
h_r	Radiative heat transfer coefficient, W/m^2K
I_T	Solar radiation incidence on the collector plane, W/m^2
J_c, J_1	Ratiosity of the collector's cover, W/m^2
J_2	Radiosity of the infrared filter, W/m2
q_{loss}	Rate of heat loss from the collector under outdoor conditions, W/m2
q^*_{loss}	Rate of heat loss from the collector under indoor conditions, W/m2
T_a, T_3	Ambient temperature ^oC
T_c, T_1	Collector's cover temperature, ^oC
T_e	Effective heat-sink temperature of the collector, ^oC
T_{fi}	Water inlet temperature, ^oC
T_{filt}, T_2	Infrared filter temperature, ^oC
T_p	Absorber temperature, ^oC
U_b	Back loss coefficient of the collector, W/m^2K
U_{Le}	Effective heat loss coefficient, W/m^2K
B	Slope of the collector, degree
E_c, E_1	Emittance of the collector's cover
E_2	Emittance of the infrared filter
	Collector efficiency
$(\tau\alpha)_e$	Transmittance-absorptance product of the collector under indoor conditions
$(\tau\alpha)_n$	Transmittance-absorptance product of the collector under outdoor conditions

Suffices

c-a	cover-surrounding ambient
c-filt	cover-infrared filter

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APPENDICES

Appendix A : Evaluation of the collector's effective temperature, T_e, for the copper collector.

$$T_e = (F_RU_{La}/F_RU_{Le})T_a + (F_RU_{Lf}/F_RU_{La})T_{filt}$$

$$F_RU_{LA}/F_RU_{Le} = (h_{c,c-a} + h_{r,c-a})/(h_{c,c-a} + h_{r,c-a} + h_{r,c-filt})$$

$$F_RU_{Lf}/F_RU_{Le} = (h_{r,c-filt})/(h_{c,c-a} + h_{r,c-a} + h_{r,c-filt})$$

$$h_{c,c-a} = 2.8 + 3V \; ; \; V = 3 \text{ m/s} \; ; \; h_{c,c-a} = 11.8 \text{ W/m}^2\text{K}$$

An enclosure for radiation exchange consists of 3 surfaces :

1-collector's cover, 2-infrared filter and 3-surrounding ambient which is assumed to be blackbody.

The radiation exchanges at the collector's cover and the infrared filter can be written as :

$$A_1\varepsilon_1(E_{b1}-J_1)/(1-\varepsilon_1) + A_1F_{1-2}(J_2-J_1) + A_1F_{1-3}(E_{b3}-J_1) = 0$$

$$A_2\ _2(E_{b2} - J_2)/(1-\varepsilon_2) + A_1F_{1-2}(J_1-J_2) + A_2F_{2-3}(E_{b3}-J_2) = 0$$

$\varepsilon_1 = 0.88$, $A_1 = 1.318 \times 1.09\ m^2$

$\varepsilon_2 = 0.88$, $A_2 = 1.95 \times 1.95\ m^2$

$F_{1-2} = 0.89$, $F_{1-3} = 0.11$

$F_{2-1} = 0.34$, $F_{2-3} = 0.66$

$T_1 = 43.9^\circ C$ $E_{b1} = 5.67 \times 10^{-8}(T_1 + 273)^4 = 571.84\ W/m^2$

$T_2 = 39.1^\circ C$ $E_{b2} = 5.67 \times 10^{-8}(T_2 + 273)^4 = 538.12\ W/m^2$

$T_3 = 35^\circ C$ $E_{b3} = 5.67 \times 10^{-8}(T_3 + 273)^4 = 510.25\ W/m^2$

$J_1 = 567.32\ W/m^2$ $J_2 = 537.09\ W/m^2$

$h_{r,1-2} = h_{r,c-filt} = A_1F_{1-2}(J_{1-2}-J_2)/[(A_1(T_1-T_2)] = 5.605\ W/m^2K$

$h_{r,1-3} = h_{r,c-a} = A_1F_{1-3}(J_1-E_{b3})/[A_1(T_1-T_3)] = 0.705\ W/m^2K$

$F_RU_{La}/F_RU_{Le} = (11.8 + 0.705)/(11.8 + 0.705 + 5.605) = 0.69$

$F_RU_{Lf}/F_RU_{Le} = 5.605/(11.8 + 0.705 + 5.605) = 0.31$

$T_e = 0.69 \times 35 + 0.31 \times 39.1 = 36.3^\circ C$

APPENDIX B Evaluation of thermal loss from the copper collector

Indoor conditions : $T_c = 43.9^\circ C$, $T_a = 35^\circ C$, $T_e = 36.3^\circ C$

$$\begin{aligned} q^*_{loss,top} &= \varepsilon_1(E_{b1}-J_1)/(1-\varepsilon_1) + h_{c,c-a}(T_c-T_a) \\ &= 0.88(571.84 - 567.3174)/(1-0.88) + 11.8(43.9 - 35) \\ &= 138.1857\ W/m^2 \end{aligned}$$

$q^*_{back} = k(T_p-T_a)/x$; x = insulation thickness = 0.0508 m

k = insulation thermal conductivity

= 0.035 W/m K)

$= 0.035(63.7 - 35)/0.0508 = 22.03\ W/m^2$

$q^*_{loss} = 138.1857 + 22.03 = 160.22\ W/m^2$

When the surrounding ambient is blackbody at 34°C,

$$
\begin{aligned}
q_{loss,top} &= \varepsilon_1 \tau[(T_c + 273)^4 - (T_a + 273)^4] + h_{c,c-a}(T_c - T_a) \\
&= 0.88 \times 5.67 \times 10^{-8}[(43.9+273)^4 - (34+273)^4] + 11.8\ (43.9-34) \\
&= 176.82\ W/m^2 \\
q_{back} &= 0.039(63.7 - 34)/0.0508 = 22.8\ W/m^2 \\
q_{loss} &= 176.82 + 22.8 = 199.62\ W/m^2
\end{aligned}
$$