Thermal Performance of Solar Collectors with EPDM Absorber Plates

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Abstract

An experimental study of flat-plate solar collectors using ethylene, propylene diene monomer (EPDM) absorber plates is described. In spite of the high thermal resistance of this material the performance is found to compare well with metal absorbers and to be in agreement with the Hottel–Whillier–Bliss equation. There is, however, an observed increase in the heat loss coefficient for mass flow rates below a critical value.

1. Introduction

The use of extruded EPDM plastic absorber mats in solar collectors is increasing due to the moderate cost and ease of installation. These collectors perform well, particularly in the temperature range required for swimming pool heating, but there is little or no published information on their characteristics or performance.

The thermal performance of plastic absorbers does not follow obviously from metal tube and fin absorber plate theory (Hottel and Whillier 1958), since EPDM has a much lower thermal conductivity than copper or aluminium—by a factor of 1000. This fact motivated the study described here.

2. Theory

The thermal efficiency η of a solar collector is defined as the ratio of useful heat collected to the total energy available as

$$\eta = \dot{m}C_p(T_o - T_i)/G, \tag{1}$$

where $\dot{m}$ is the mass flow rate of heat transfer fluid per unit collector area ($\mathrm{kg\,m^{-2}\,s^{-2}}$), C_p is the specific heat of the transfer fluid ($\mathrm{J\,kg^{-1}\,K^{-1}}$), T_i and T_o are the inlet and outlet temperatures of the heat transfer fluid (K) respectively, and G is the total solar radiation flux incident on the collector surface ($\mathrm{W\,m^{-2}}$).

The Hottel–Whillier–Bliss equation (Bliss 1959) expresses η as a function of T_i, T_o, $\dot{m}$, and the sun zenith angle relative to the collector surface θ, as

$$\eta = F'(Kt\alpha - U\chi_f) \tag{2}$$

$$= F'F''(K\tau\alpha - U\chi_i), \tag{3}$$

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where F' is the efficiency factor (dimensionless), K is the incident angle modifier (dimensionless), given by

$$K = 1 - b(1/\cos\theta - 1), \tag{4}$$

$\tau\alpha$ is the transmission absorptance coefficient (dimensionless), U is the heat loss coefficient ($\mathrm{W\,m^{-2}\,K^{-1}}$), and the flow factor (dimensionless) is

$$F'' = \mu\{1 - \exp(-1/\mu)\}; \qquad \mu = \dot{m}C_p/F'U. \tag{5}$$

Further, b is the incident angle modifier coefficient (dimensionless), $\chi_f = (T_f - T_e)/G$, $\chi_i = (T_i - T_e)/G$, T_f is the mean heat transfer fluid temperature (K), and T_e is the equivalent environmental temperature (K).

Using equations (2)–(5), the value of η obtained by measuring all the quantities on the right side of (1) can be correlated with the operator controlled parameters ($\dot{m}, \theta, T_i$) and the environmental parameters (G, T_e). The resultant correlation coefficients (F', F'', b, U) then provide a comparative measure of the thermal performance of different solar collectors.

3. Equipment

All solar collector tests were performed using an outdoor test facility, located on the roof of the Physics Department at Monash University. This facility can test four $2\times1\ \mathrm{m^2}$ solar collectors simultaneously, and the solar collector stand has a zenith and azimuth angle pivot which enables the solar collectors to be oriented at normal incidence to the sun ($\theta = 0°$) for four hours either side of solar noon. The water inlet temperature to the solar collectors is maintained by water immersion heaters which are controlled by a Hewlett Packard data acquisition system to within 0·10 K of the required value. The mass flow rate to each solar collector is supplied by a constant head water tank and can be adjusted as required by a control valve on each outlet line. The range of operation and measurement error of each parameter is summarised in Table 1.

Table 1. Allowed variation of measured parameters and associated instrument errors

Parameter	G ($\mathrm{W\,m^{-2}}$)	T_i-273 (K)	T_o-273 (K)	T_e-273 (K)	$\dot{m}$ ($\mathrm{kg\,m^{-2}\,s^{-1}}$)
Range of measurement	700–1100	10–60	10–60	10–35	0·01–0·03
Instrument error	±3%	±0·1	±0·1	±0·1	±0·3%
Allowed variation during a test	±10	0·5	0·5	1·0	±0·5%

4. Experimental Method

The results reported in this paper were all obtained in accordance with the ASHRAE standard 93-77 (1977) and the Australian Standard AS2535 (1982). When the environmental conditions were suitable for solar collector testing, $\dot{m}$, T_i and θ were set to the required values and the parameters G, T_i, T_o and T_e were monitored at ten-second intervals until their variation was less than that specified in Table 1 over the test period. Two additional requirements were imposed as criteria for a valid

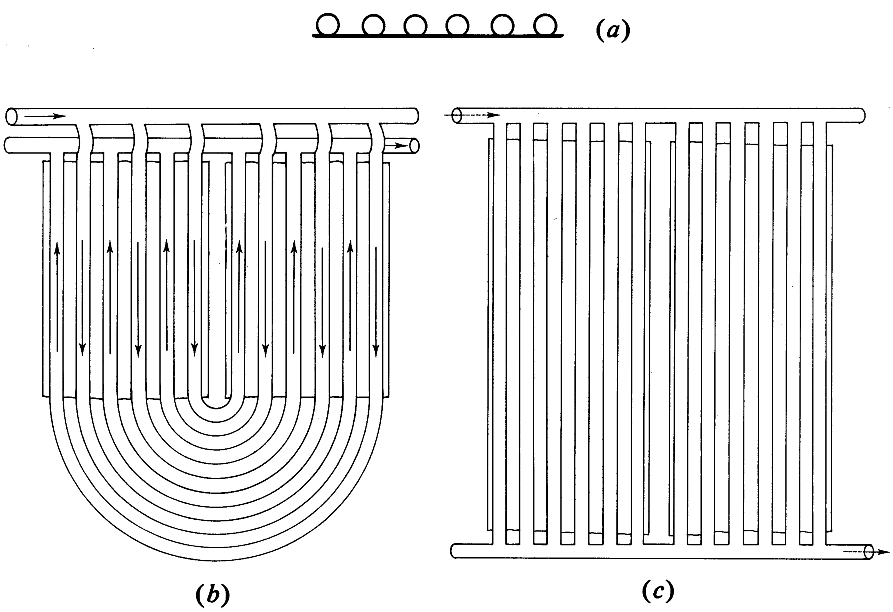

Fig. 1. (*a*) Cross section of a typical EPDM absorber mat. (*b*) Two mats linked to headers to form a solar collector in which adjacent risers have flow in opposite directions. (*c*) A more conventional collector employing two EPDM mats. For thermosyphon operation the inlet is at the bottom and the outlet at the top.

thermal efficiency measurement: The wind speed V_w must be less than $1 \cdot 5 \ \mathrm{m\,s^{-1}}$ and the variation in the thermal efficiency must be less than $0 \cdot 02$ over the test period. Since the test period specified by AS2535 (1982) and ASHRAE 93-77 (1977) is always less than ten minutes for the range of $\dot{m}$ considered, a constant test period of ten minutes was used for all tests.

Some 15 collectors were assembled and tested, all collectors using EPDM mats to form the collector plates. These mats were manufactured by a continuous strip extrusion process and are comprised of a flat sheet about 10 cm wide and about 2 mm in thickness. On one side tubes (the risers) are formed of about 2 mm wall thickness and 1 cm outer diameter fixed at about 2 cm between centres, while the tubes run longitudinally on the flat sheet and the whole assembly comes in a long roll which may be cut easily to any convenient length. One working collector on the roof of the Monash University swimming pool has mats which are 32 m long. Different manufacturers produce mats of slightly different tube size and spacing from those given here, but these details are not important. Fig. 1 shows a cross section of a typical mat and two possible arrangements of mats to form a collector. The mats are light and very flexible and are easily connected to inlet and outlet manifolds. The arrangement of Fig. 1*b* was thought to give a higher efficiency than the more conventional collector in Fig. 1*c*. All collectors were assembled in containers with back and edge insulation and with transparent top covers.

5. Results

Some 850 thermal efficiency measurements were made on collectors operating outdoors over a three-year period. No differences were found in the performance of collectors employing EPDM mats of the three types available, so that within fairly narrow limits the tube sizes and spacings are not critical. The arrangement of Fig. 1*b* should be a better collector than that of Fig. 1*c*, since it should result in a more even temperature distribution over the collector plate and hence a reduced heat loss. The results indicate a possible reduction in heat loss, but since this reduction fell within the experimental error of the measurements this conclusion cannot be made. However, in large roof mounted collectors there may be practical advantages in installing the inlet and return manifolds, either as in Fig. 1*b* or 1*c*, and either may be used with little change in the end result.

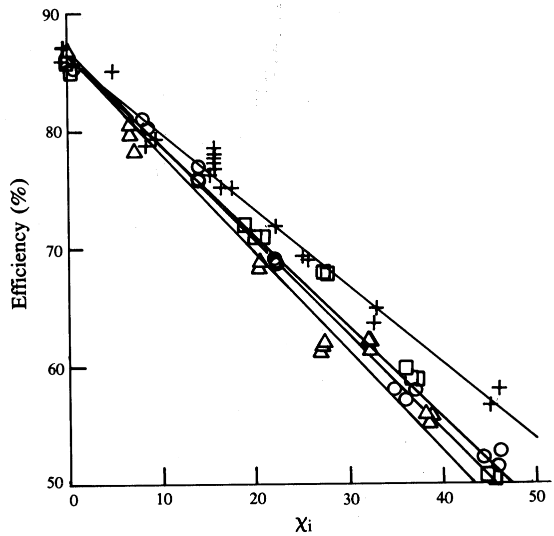

Fig. 2. Efficiency of an EPDM collector as a function of $\chi_i = (T_i - T_e)/G$ for four different flow rates: crosses, $F'' = 0\cdot94$; squares, $F'' = 0\cdot93$; circles, $F'' = 0\cdot91$; and triangles, $F'' = 0\cdot84$.

Fig. 2 shows a typical set of efficiency curves for an EPDM collector at four different mass flow rates. It can be seen that the results are in accord with the theory (see equations 2 and 3). The gradient of the curves gives the heat loss coefficient and the result here of about $7\cdot0\ \mathrm{W\,m^{-2}\,K^{-1}}$ is similar to that for single-glazed metal collectors (Duffie and Beckman 1980). The heat loss is independent of temperature but shows a dependence on mass flow rate, as discussed below. The similarity to metal collectors is quite remarkable. For example, the incident angle modifier coefficient found for the EPDM collectors was $b = 0\cdot09$ in accord with that predicted for metal collectors of similar geometry (Duffie and Beckman 1980). The explanation lies in the fact that although EPDM has a poor thermal conductance the heat path to the collector fluid is still of much lower resistance than the heat loss path. This heat loss is mainly due to radiation which is inhibited by the glazing.

The variation of the heat loss coefficient with the flow factor is shown in Fig. 3 for three different collectors. The sharp drop in the heat loss coefficient at flow factors of about $0\cdot95$ is difficult to explain and this feature is not present with all-metal collectors. While a steady drop in heat loss with increasing mass flow is to be expected, there is at present no mechanism to explain the sharp drop at one particular

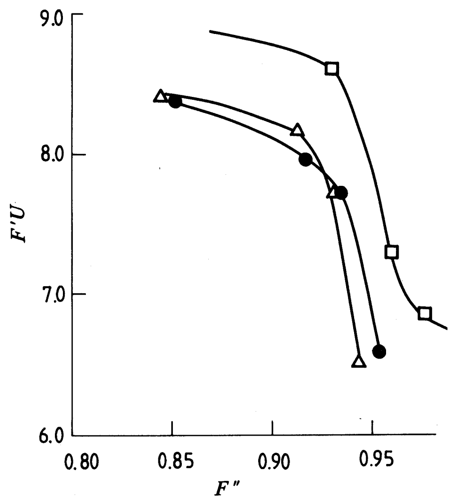

Fig. 3. Heat loss $F'U$ as a function of the flow factor F'' for three different EPDM collectors.

flow value. The drop is not due to a transition from laminar to turbulent flow of the heat transfer fluid. The Reynolds number R_e for the flows reported here is typically about 500. The transition to turbulent flow would occur for $R_e > 2000$.

The explanation given here arises out of a concurrent investigation (O'Keefe and Francey 1986) into the flow distribution within the riser tubes of the collector. It was noted that at flow factors smaller than 0·95 there was air present in the riser tubes—some of these were transparent for investigation of the flows. The air of course formed pockets above the fluid and immediately under that part of the riser facing incoming solar radiation. Thus, the top part of the riser will move to a temperature higher than parts further round a circumference, the high resistance of the EPDM supporting the thermal gradient. The higher temperature leads to a higher heat loss. Metal risers on the other hand are isothermal. At flow factors above 0·95 the air pockets disappear but reform when the flow factor is again reduced. A flow factor of 0·95 corresponds to a flow rate of about 15 $g\,s^{-1}\,m^{-1}$ and care must be taken to maintain flow rates above this if the best results are to be obtained from these collectors.

As mentioned above, a large collector 480 m^2 in area is operating on the roof of the Monash University swimming pool and this collector was one of the 15 tested. The performance of this collector has been reported elsewhere (Francey *et al.* 1985), but a small (2·0 m^2) section of the large collector was tested independently in an effort to predict the performance characteristics of such large collectors. The roof top collector has EPDM mats glued directly to the metal roof but sitting in depressions between metal ribs over which is placed the acrylic sheet glazing. This gives a plate to top cover gap of 4 cm and results in some shading of the plate. This gave an incident angle modifier coefficient of 0·172 for the pool collector. The small scale model, which also had 4 cm side walls, gave a coefficient of 0·169. The optical efficiency ($F'K\tau\alpha$) for the large collector was 0·74, while for the small model this was 0·76. Thus, excellent agreement was reached between the two.

6. Conclusions

In spite of the high thermal resistance of EPDM, solar collectors using flat plates and risers formed from this material give results comparable with all-metal tube and fin collectors. Care must be taken to maintain an adequate flow of heat transfer fluid within EPDM collectors. Scale models of large solar collector installations can predict their expected performance.

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