The efficient collection and long term storage of solar energy in the UK, using air as the working fluid

Thesis

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Tadeusz Oreszczyn

B.Sc. (Hons) Brunel

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CHAPTER 5 Testing Air Heating Collectors

Solar collector testing is essential to allow the technical comparison of different collectors and to predict the operation of the collector under different conditions.

Most test methods involve characterising the collector by the Hottel-Whillier steady state equation based on an energy balance of the absorber plate.

\[ Q_u = F_R A_C \left[ I_m (\tau a)_e - U_L (T_i - T_a) \right] \]  

(5.1)

The efficiency is given by

\[ \eta = \frac{Q_u}{I_m A_C} = F_R (\tau a)_e - F_R U_L \frac{(T_i - T_a)}{I_m} \]  

(5.2)

\( F_R (\tau a)_e \) and \( F_R U_L \) are usually considered to be independent of temperature.

The efficiency of the collector can be measured for different inlet temperatures by measuring the mass flow rate and the temperature rise of the air

\[ \eta = \frac{Q_u}{A_C I_m} = \frac{mc_p (T_e - T_i)}{A_C I_m} \]  

(5.3)

Equation 5.2 indicates that if the efficiency \( \eta \) is plotted against \( (T_i - T_a)/I_m \) a straight line will result where the slope is \( F_R U_L \) and the y intercept is equal to \( F_R (\tau a)_e \). In
reality, $U_L$ depends on the mass flow rate, the temperature of the collector, and the ambient weather conditions. In addition $(\tau\alpha)_e$ varies with the incident angle between the solar beam and the collector or some effective angle in diffuse conditions, and even for a given $m$, $F_R$ depends on $U_L$ and the heat transfer in the duct. The various test methods attempt to control the test conditions so that a well-defined efficiency curve can be obtained.

In the mid 1970's many new collector designs appeared on the commercial market. A need developed for standard tests to produce data of the type required in process design. In response, the US National Bureau of Standards devised a test procedure [1] which has been modified by ASHRAE, to form the ASHRAE 93-77 standard procedure [2] and a similar method has been adopted in Europe by the Commission of the European Communities [3]. The standard covers indoor and outdoor measurements. The method involves measuring the efficiency using equation 5.3. The outdoor measurements (as well as indoor) must be carried out under strict conditions of pseudo steady radiation $>600$ Wm$^{-2}$. Measurements need to be performed for at least four different values of fluid inlet $T_i$. At least four data points should be taken for each value of $T_i$ integrated over a time period equal to the collector time constant: two during the time period preceding noon and two in the period following solar noon. For the outdoor measurements in the United Kingdom steady state conditions may only prevail two or three times in the year, making
testing very difficult and long winded, spread over many days. Trying to achieve true climatic conditions indoors, in particular a solar spectrum, can be very difficult and expensive, and explains why only one such facility exists in the United Kingdom - at the Solar Energy Unit Cardiff [4].

In 1976 the EEC started a 'testing programme' with the aim of proposing a limited number of test methods, which are suitable for European climatic conditions. It is intended that they should be compared with each other, then the reproducability of ASHRAE testing under different conditions is examined [5] and that the number of days required to achieve an efficiency curve is reduced to almost any 2 or 3 clear sunny days. They suggest performing tests in a 'monotonous' transient to allow diurnal variations. The useful energy collected is then

\[(Q_u)_t = Q_u - (mc)_e \frac{dT_f}{dt}\]  

(5.4)

where \((mc)_e\) is defined as the effective heat capacity of the collector, all the mass being assumed to lie in the absorber plate, and \(Q_u\) would be the energy collected if the collector was in the steady state. Assuming that the heat transfer coefficient, \(U_L\), and \(F_R\) is the same in steady state and in transient conditions, the efficiency \(\eta\) is defined as equal to
In order to keep the term \((mc)_e \frac{dT_f}{dt}\) relatively small compared to \((Q_u)_t\) the transient is limited to \(\frac{dT_f}{dt} < 12^\circ C \, hr^{-1}\), and \(\frac{dT_f}{dt}\) must be kept constant during the duration of the test.

This method has shown to give good agreement of results with those obtained by steady state measurements but allows more than one measurement at a particular fluid inlet per day. Yet to perform a test this method still requires two or three clear days of a kind which occur infrequently in the United Kingdom. However, because of the cost of indoor testing and time for outdoor testing, a fully transient test method has been included in British Standard DD77:1982[6]. This allows the pseudo-steady state properties to be calculated from integrated data for incident energy and collected energy for various fixed inlet temperatures, the insolation being allowed to vary in any manner. The efficiency of the collector is then calculated using not only the insulation over the integrated period but also the insulation of the previous integrated periods, up to two time constants earlier. As yet this method has not been tested on air heating collectors, although it appears to work well for water heating collectors. Section 5.3.2 of this chapter examines the usefulness of this test method for air heating collectors.
To minimise the problems of obtaining the correct incident radiation, be it indoors or outdoors, a 'zero radiation heat loss' test [7] has been devised, as follows.

If the collector is shielded from radiation \((I = 0)\) by the use of a screen but supplied with pre-heated fluid, from equation 5.1 the collector heat gain is

\[
Q_u = -F_R U_{AC} (T_i - T_a)
\]  

(5.6)

thus enabling the measurement of \(F_R U_L\) to be made.

Furthermore, if the collector is exposed to radiation but with the inlet heat transfer fluid at ambient temperature equation 5.1 becomes

\[
Q_u = F_R A I (\tau \alpha)_0
\]  

(5.7)

So \(F_R(\tau \alpha)_0\) can be measured.

Although this method works well for water heating collectors, doubt over its applicability to rear-duct air heating collectors remains, because of an inversion in the duct temperature gradient [8]. Section 5.5 examines the use of this test method for air heating collectors.

The two most applicable test methods for the UK climate, namely transient and 'zero radiation heat loss' have not been verified for air heating collectors. The rest of this chapter therefore examines the testing of two air heating collectors by the ASHRAE standard, transient testing and zero
radiation, and compares the results. It also examines the usefulness of testing air heating collectors with a cheap low intensity solar simulator. It then goes on to examine if the results produced from standard steady state collector testing provide any useful information about the operation of solar air collectors in the United Kingdom, where only 10% of the solar energy falls with an average intensity greater than 500Wm$^{-2}$ for a period of an hour (see Figure 5.1) and a large proportion of the radiation falls intermittently.
5.1 Collector types tested

All the collector testing has been performed on a rear duct collector made by D.C.Hall and a top duct collector made of structured polycarbonate.

5.1.1 D.C.Hall Collector

This is a rear duct air heating collector, with a polycarbonate cover, selective surface and a black finned rear duct (see Figure 5.2). The collector was designed as a high performance collector to produce an efficient yet simple to build collector which could be used to test the reduction in top loss coefficient by different gases (see Chapter 7). The collector components were designed by me, with the assistance of Dr.B.W.Jones, built by 'D.C.Hall', Aspley Guise, Bucks, and assembled at the Open University.

Materials

Cover - 3mm thick polycarbonate, refractive index 1.586, extinction coefficient 20m⁻¹, normal transmissivity 0.87, thermal radiation transmittance = 0.04. The variation of transmissivity with angle was calculated using the Fresnel equations [9], the material extinction coefficient and accounting for multiple reflection, and is plotted in Figure 5.3. The thermal emissivity was measured as 0.85.
Absorber - HS15TB aluminium alloy, thickness 0.914mm
Selective top coating - 'Maxorb' selective absorbing nickel foil manufactured by MPD Technology Limited, Birmingham. The nickel foil's selective properties result from a micro-roughened mixed nickel-chromium oxide layer. The reverse side of the blackened foil has a pressure sensitive adhesive. Care must be taken in applying the foil to the absorber because finger prints ruin the selective nature of the surface. The foil's thickness is 13 microns and comes in widths of 148mm. The foil is supplied with expansion joints to prevent wrinkling due to differential expansion with respect to the aluminium absorber plate. The manufacturer's inspection report quotes an absorptivity 0.98 at solar wavelengths and an emissivity 0.09 at 20°C for normal angles. The foil's absorbance cuts off at wavelengths > 1.0 microns [10]. The angular reduction in absorptivity for 'Maxorb' is less than 3% for a 60° angle of incidence to normal [11]. The hemispherical to normal emittance has not been measured but is assumed to be the same as most metals $E_h/E_n \approx 1.2$ [12].

Rear Duct

Base - HS 15 TB aluminium alloy, thickness 1mm, with T piece supports
Rear Duct

Coating - 'Nextel 2010' a non selective black solar absorbing paint manufactured by 3M, Bracknell. Absorbtance 0.97 (for wavelengths 0.28μ to 2.5μ). Hemispherical emittance 0.91 (for wavelengths 2.4μ to 26.0μ) [13].

Fins - Aluminium tee pieces thickness 3.5mm (Figure 5.4).

Insulation - 'Superwrap 100' 4" thick fibreglass insulation manufactured by Pilkingtons, thermal conductivity 0.04Wm\(^{-1}\)K\(^{-1}\), total insulation thickness, 390mm.

Cover Support - Acrylic thickness 20mm, with aluminium glazing bars to seal absorber and cover.

Collector Support and base - Water boiling point plywood thickness 10mm finished with 3 coatings of yacht varnish.

Edge Insulation - Styrofoam cellular polystyrene, thickness 75mm.

Sealant - All joints both in the top and rear duct were sealed with Dow Corning 'Silicone Sealant 781'
chosen for its high temperature properties and long life time [14].

**Heat Capacity of the components in a D.C. Hall Collector**

*(aperture 0.857m²)*

<table>
<thead>
<tr>
<th>Volume</th>
<th>$\rho C_p$</th>
<th>Heat Capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>/m³</td>
<td>/MJm⁻³K⁻¹</td>
<td>/kJK⁻¹</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Component</th>
<th>$V$</th>
<th>$\rho C_p$</th>
<th>$C$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cover, polycarbonate</td>
<td>17.8 $\times 10^{-4}$</td>
<td>1.44</td>
<td>2.56</td>
</tr>
<tr>
<td>Cover support, acrylic</td>
<td>35.0 $\times 10^{-4}$</td>
<td>1.80</td>
<td>6.30</td>
</tr>
<tr>
<td>Glazing bars, aluminium</td>
<td>11.4 $\times 10^{-4}$</td>
<td>2.45</td>
<td>2.79</td>
</tr>
<tr>
<td>Absorber, aluminium</td>
<td>8.14 $\times 10^{-4}$</td>
<td>2.45</td>
<td>1.99</td>
</tr>
<tr>
<td>Fins, aluminium</td>
<td>4.33 $\times 10^{-4}$</td>
<td>2.45</td>
<td>1.06</td>
</tr>
<tr>
<td>Rear duct base</td>
<td>24.6 $\times 10^{-4}$</td>
<td>2.45</td>
<td>6.02</td>
</tr>
<tr>
<td>Bottom insulation, fibreglass</td>
<td>0.4</td>
<td>0.033</td>
<td>13.2</td>
</tr>
<tr>
<td>Support frame wood</td>
<td>0.023</td>
<td>0.708</td>
<td>16.3</td>
</tr>
<tr>
<td>Base, wood</td>
<td>0.012</td>
<td>0.708</td>
<td><strong>8.55</strong></td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td></td>
<td><strong>58.8kJK⁻¹</strong></td>
</tr>
</tbody>
</table>

*Note* that the capacitances are only additive if they are (in my terms) all in parallel and with no resistance between.
5.1.2 Structured Polycarbonate (S.P) Collector

This is a top duct air heating collector, made from structured polycarbonate, the absorbing surface is painted with matt black paint (see Figure 5.5 and Plate 5.1). The collector was designed for easy construction, cheapness, small heat capacity (patent pending [15]). The collector was built and constructed at the Open University. This design was originally thought to be novel. However, a subsequent investigation revealed a similar design already patented by Erb [16][17], nullifying any similar patents.

Materials

Top duct - Double skin structured polycarbonate, 'Makrolon SDP lightweight' manufactured by Roehm Ltd., London.

Weight 2.7kg per m². See Figure 5.6 for dimensions. Normal transmissivity ≈ 80%. The angular variation of transmissivity for a similar structure is shown in Figure 5.7. The measured thermal emissivity is 0.85.

Absorber Coating - 'Nextel 2010' non selective black solar absorbing paint, absorbtance 0.97.[13].

Insulation - Fibreglass 7cm batt insulation, thermal conductivity 0.04 Wm⁻²K⁻¹.
Support Structure - Wood coated on outside with 2 coats of yacht varnish.

Edge Insulation - Fibreglass 5cm batt insulation, thermal conductivity $0.04 \text{ Wm}^{-2}\text{K}^{-1}$.

Heat Capacity of the components in a SP collector (aperture $1 \text{ m}^2$).

<table>
<thead>
<tr>
<th>Volume/m$^3$</th>
<th>$\rho C_p$ /MJm$^{-3}\text{K}^{-1}$</th>
<th>Heat capacity /kJk$^{-1}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top duct, polycarbonate $24 \times 10^{-4}$</td>
<td>1.44</td>
<td>3.39</td>
</tr>
<tr>
<td>Bottom insulation fibreglass 0.07</td>
<td>0.033</td>
<td>2.31</td>
</tr>
<tr>
<td>Edge insulation fibreglass 0.028</td>
<td>0.033</td>
<td>0.92</td>
</tr>
<tr>
<td>Support structure, wood $5.6 \times 10^{-4}$</td>
<td>0.78</td>
<td>0.40</td>
</tr>
</tbody>
</table>

Total 7.02 kJk$^{-1}$. 
5.2 Instrumentation

Air collector tests involve measuring the following parameters:

- temperature (air inlet, air outlet, ambient, absorber)
- mass flow rate,
- wind speed,
- insolation, direct and diffuse,
- pressure difference across duct,
- humidity.

These parameters need measurements over a long period of time and some need integrating, therefore the data were collected with the help of a data logger. The following instrumentation was used to measure the above parameters.

**Insolation** - Kipp and Zonan solarimeter calibrated to the international pyrheliometer scale 1956. Since only one solarimeter was available for outdoor testing this measured the total diffuse and direct radiation, and spot checks were carried out for the diffuse radiation by screening out the direct radiation using a 9cm diameter silvered disc held half a meter above the Kipp and Zonan. A similar instrument was used indoors.

**Wind speed** - A cup anemometer Ref IM124 (Met ref.992) supplied by R.W.Munro Ltd., London was used for outdoor
testing. This has a threshold of 0.45 ms$^{-1}$ but can measure wind speeds up to 46 ms$^{-1}$. Indoors a hot wire anemometer (AVM 501 TC Prosser Scientific Ltd.) was used.

**Temperature** – Platinum resistance thermometers meeting British Standard 1904:1964, Grade II [18] (that is 100° at ice point ± 0.1 %) were used. Each resistor is 31mm long 3mm diameter and was manufactured by Mattey Printed Products Ltd., Stoke on Trent. Ambient air temperatures were measured outdoors in a white painted Stevenson screen and in a metal screen indoors.

**Mass flow rate** – Orifice plates constructed in accordance with British Standard 1042 [19] were used both indoors and outdoors (see Figure 5.8). The pressure difference across the orifice plate was monitored by a 'Type FC040 Furness Control Ltd., pressure transducer' (0 to 100 mm H$_2$O = 0 to 1V). The system was calibrated against a calibrated 'Periflow' Orifice plate (Ref 1858) – too expensive to use in the test rig. The pressure drop across the Periflow orifice during calibration was monitored with an 'Airflow type 504' manometer 0 to 12 mm, which had been levelled.

**Pressure difference** across the collector both indoors and outdoors was monitored with four external manifold pressure taps 1.6 mm in diameter, on the inlet and outlet of the collector. The inlet pressure taps were half a pipe diameter from the collector inlet and two pipe diameters from the
collector outlet. The pressure difference between the inlet and outlet was measured with a levelled 'Airflow type 504' manometer.

**Humidity** - This was monitored by means of a Darton combined thermo-hygrograph calibrated by means of a whirling hygrometer.

**Data logger** - A 'Microdata M1600 with 20 channels was used, scanning every 1 minute. Each transducer has its own separate interface card, which was calibrated in the laboratory. Insolation, mass flow rate and wind speed values were all integrated during the scan period. The data logger also recorded the date and time of each scan. The data were then stored on a 'Verbatim data cartridge'.
5.3 Outdoor testing

5.3.1 Steady state testing according to ASHRAE

Figure 5.9 shows the testing configuration recommended by ASHRAE standard 93-77 [2]. This was replicated as closely as possible with the available equipment, Figure 5.10. The following differences are to be noted.

(i) No air reconditioning apparatus was available. Instead the air was only passed once through the collector, but it could be pre-heated by a 1kW bar heating-element.

(ii) the collector had to be mounted horizontally on the roof of the science faculty laboratories at the Open University campus, Milton Keynes. This meant that the angle of incidence of the sun to the normal of the collector had to be greater than the recommended maximum angle of incidence of 30° for some of the tests. This was compensated by an angle of incidence modifier. The roof also happened to be painted white and the collector had to be placed within 2 meters of a 1m parapet. This made evening tests impossible because the collector was then shaded.

(iii) The temperature difference of the fluid inlet and outlet was measured by calibrated platinum resistance
thermometers (PRTs) connected in two arms of a resistance bridge. This method is not that which is recommended. Instead, the use of a thermopile is suggested. No problems were recorded using PRTs and an accuracy of ± 0.2°C was achieved. Given the high temperature rise associated with air collectors this level of accuracy is permitted (see Section 5.5). No mixing devices were felt necessary because the desired temperature uniformity of ± 0.5°C across the duct profile was achieved.

(iv) Air flow measurements were made with an orifice plate, not a nozzle as recommended, because the collector area, and hence the mass flow rate, was small enough to necessitate calibration for a nozzle and so no benefit would have occurred from using a nozzle. Indeed the construction of a nozzle is far more complicated than for an orifice.

**Collector time constants**

Before steady state testing can be carried out some knowledge is required of the time the collector requires to reach a steady state, so that the proper time intervals for the quasi-steady state efficiency test can be used.

The governing equation for the transient behaviour of a solar collector, assuming a one nodal model, is
Provided \((\tau\alpha)_o\), \(U_L\), \(T_a\), \(m\) and \(c_p\) are considered constant over the transient period and if the rate of change of the transfer fluid exit temperature with time is related to the rate of change of transfer fluid average temperature with time by

\[
\frac{dT_f}{dt} = k \frac{dT_e}{dt} \tag{5.9}
\]

then Equation 5.8 can be solved to give the exit temperature of the transfer fluid as a function of time:

\[
\frac{(mc)_e}{A_c} \frac{dT_f}{dt} = P_L \left( (\tau\alpha)_o - P_R \frac{U_L (T_i - T_a)}{T_e - T_i} - \frac{mc}{A_c} (T_e - T_i) \right) \tag{5.8}
\]

\[
\frac{P_R I(\tau\alpha)_o - P_R U_L (T_i - T_a) - (mc)/A_c (T_e, t - T_i)}{P_R I(\tau\alpha)_o - P_R U_L (T_i - T_a) - (mc)/A_c (T_e, initial - T_i)}
\]

\[
e^{- \left[ \frac{mc_p}{k(mc)_e} \right] t} \tag{5.10}
\]

where

\[
k = \frac{mc_p}{P' U_L A_c} \left[ \frac{P'}{P_R} - 1 \right] \tag{5.11}
\]

The quantity \(k(mc)_e/mc_p\) is known as the time constant and is the time required for the quantity on the left side of equation 5.10 to change from 1.0 to 0.368, where 0.368 \(\approx 1/e\).

The ASHRAE outdoor standard recommends measuring the time constant by keeping (i) the inlet fluid to within \(\pm 1^\circ C\) of the ambient; (ii) a steady mass flow rate; (iii) quasi-
steady state conditions with an incident flux of greater than 790 Wm\(^{-2}\). The collector is then shielded from radiation and the time is measured for the fluid exit temperature to reach 0.368 of the difference between the initial fluid exit temperature and the fluid inlet temperature.

Figure 5.11 shows the cooling curve of the structured polycarbonate collector. The collector never reaches ambient temperature because the collector was screened with a white polystyrene sheet slightly larger than the aperture and so some diffuse radiation could strike the collector and some direct radiation could reflect off the floor to the screen and hence reach the collector. There will also be slight warning of the fluid due to friction in the system. To adjust for this the time constant corresponds to when

\[
\frac{T_e - T_{e, \text{final}}}{T_{e, \text{initial}} - T_{e, \text{final}}} = 0.368
\]  

(5.12)

The validity of this method of measuring the collector time constant has been questioned by G.R. Mather [20].

Collector thermal efficiency

The efficiency of the flat plate collector operating in the steady state for normal beam irradiance is given by
We assume that the aperture area is the gross collector area, which for experimental developmental collectors is a fair assumption.

By performing efficiency tests only during particular times and under particular weather conditions an efficiency curve can be obtained with a minimum of scatter.

The ASHRAE recommended constraints for taking measurements are these:-

that at least four different values of inlet fluid temperature shall be used, and at least four data points shall be taken at each fluid inlet temperature; two during the period preceding solar noon and two in the period following solar noon. Data points shall be determined by integrating the data over a time period equal to the time constant of the collector, or 5 minutes, whichever is the longer. For air collectors this may be up to 15 minutes [21]. In conducting these tests, care should be taken to ensure the following.

(i) Incident solar radiation is steady for each time interval during which an efficiency value is calculated.

(ii) The transfer fluid should be circulated through the collector at the appropriate inlet temperature level.
until the temperature has remained constant for 15 minutes prior to the period in which data will be collected.

(iii) The wind velocity across the collector shall remain steady and less than $4.5 \text{ ms}^{-1}$.

(iv) The tests shall be carried out at times when the average irradiation shall not be less than $630 \text{ Wm}^{-2}$.

Obtaining all these conditions pose a very serious problem in the United Kingdom as four days during which the sun shines uninterrupted for four hours around noon are required for each efficiency test. The measurement of one such efficiency test was reported by Wiles [8] as having taken half a year, and the Commission of European Communities in their test recommendations [3] say 'As a result of these restrictions, tests in most of central and Northern Europe often take several months to complete because suitable climatic conditions occur only rarely.'

Figure 5.12 shows an example of uninterrupted radiation as defined in the ASHRAE standard [2]. Radiation as constant as this in the UK is very rare. The most constant radiation measured in the summer of '83 (a summer of above average insolation for the UK) is shown in Figure 5.13. Usually, even for comparatively clear skies, there are fluctuations in radiation, including around noon.
Incident angle modifiers

Equation 5.13 is only true for radiation incident normal to the collector. To allow collector testing for angles other than normal an equivalent normal solar irradiance must be calculated, this is defined as the irradiance normal to the plane of the aperture whose effect on the thermal output of the collector is calculated to be the same as that of the actual total solar irradiance. To convert from a measured diffuse ($I_d$) and direct ($I_b$) radiation to the equivalent normal irradiation ($I$) the Draft British Standard [6] recommends the use of three correction factors $F_1$, $F_2$ and $F_3$.

$$ I = F_1 F_2 I_b + F_3 I_d \quad (5.14) $$

$F_1$ is the correction due to the shading of the absorber plate from direct radiation by the sides of the collector - the standard neglects any reflectance from sides. $F_2$ corrects for the variation in optical properties of absorbtance and transmission with angle of the cover and absorber for direct radiation. $F_3$ corrects for the variation in optical properties for diffuse irradiation. No correction for shading of diffuse irradiation by the collector sides is recommended, because the effect is negligible.

Data were obtained over six days at the Open University to establish an efficiency curve for the D.C.Hall collector (Section 5.1.1). Figure 5.14 shows the weather data (insolation and wind) for the test period. Measurements were made between 11.00 hours and 15.00 hours. Figure 5.15 shows
the angle of incidence of the direct radiation on the (horizontal) cover of the collector during the experimenting period, which varies between 30° and 45° to the normal. The equivalent normal solar irradiance was calculated using the procedure suggested in the Draft British Standard [6]. Because \( P_1 \) and \( F_2 \) are only weak functions of angle between 0° and 50° (see Figure 5.16) the values of \( P_1 \) and \( F_2 \) corresponding to 35° (the average angle of incidence during experimentation) has been used in all calculations. It has also been assumed that a constant proportion of diffuse irradiation is present. So equation 5.14 reduces to

\[
I = 0.93 I_m
\]  

(5.15)

The reduction in intensity of 0.93 is principally due to over shading; which is largely due to the collectors small aspect ratio.

For the structured polycarbonate collector

\[
I = 0.94 I_m
\]  

(5.16)

Efficiency errors

The errors are: ± 3% error in mass flow rate measurement; ± 0.3°C in air temperature difference; ± 4% in insolation [22]; and negligible errors in the specific heat capacity of air. The combined standard measurement error for each efficiency point is then ± 6% for the maximum temperature differences, and ± 10% for the minimum temperature differences. These errors are not far from Aranovitchs conclusion [22] that the accuracy of efficiency curves under
the best conditions is in the range of $\pm 6\%$ to $\pm 8\%$.

The ASHRAE standard does not lay down any limitations on acceptable air leakage rates in collectors, or how such leaks should be detected. But they are important.

**Leakage**

Air leaks in solar collector testing are very difficult to detect, particularly under the test conditions of negative pressure, because smoke or tracer gas detection is impossible. So great care is needed in both constructing the collector and in connecting the duct work, to eliminate any such leaks between the point where $T_{in}$ and $m$ are measured. Leaks anywhere else do not affect the measured result. Even small leaks can cause large errors. For example let's investigate the effect of a leak at the connection between the inlet temperature sensor and the collector (Figure 5.17(a)) of ambient air at a leakage flow rate of $m_l$.

\[
T_i = \frac{T_a m_l + T_{im} (m - m_l)}{m}
\]

(5.17)

where $T_{im}$ is the temperature of the fluid measured at the fluid inlet temperature sensor.

The true efficiency of the collector is

\[
\frac{mc_p}{A} (T_e - T_i) = IF_R (\tau a) - F_R U_L (T_i - T_a)
\]

(5.18)

but the measured efficiency is
\[
\frac{mc_p}{A} (T_e - T_{im}) = I \left[F_R(\tau\alpha)\right]_m - \left[F_{RLU}\right]_m (T_{im} - T_a) \tag{5.19}
\]

Where the subscript \( m \) refers to the measured value.

When the fluid inlet temperature is ambient both \( T_i \) and \( T_{im} \) are equal to ambient. So, comparing equations 5.18 and 5.19 we see that \((F_R \tau\alpha) = (F_R \tau\alpha)_m\). This means that if a leak is present at the inlet the correct value of \( F_R (\tau\alpha) \) will still be measured.

So substituting \( I F_R(\tau\alpha) \) from equation 5.18 into \( I[F_R(\tau\alpha)]_m \) of equation 5.19 and substituting for \( T_i \) from 5.17 we get

\[
F_{RLU} = \frac{\left[F_{RLU}\right]_m - \left[\frac{c_p m_i}{A}\right]}{\left[1 - \frac{m_i}{m}\right]} \tag{5.20}
\]

Equation 5.20 enables the calculation of the true value of \( F_{RLU} \) given the measured value \( [F_{RLU}]_m \) and the leakage rate at the collector inlet. Since there are no temperature terms in equation 5.20, if there is an air leak at the collector inlet a straight line efficiency curve will be measured with the true y axis intercept but an incorrect gradient.

Now consider air leaks between the collector outlet and the temperature sensor at the outlet (see Figure 5.17(b)). The true collector outlet temperature and the measured outlet
temperature are related by

\[ T_{\text{em}} = \frac{T_a m_l + T_e (m - m_l)}{m} \]  
(5.21)

The true efficiency of the collector is

\[ (m - m_l) \frac{c_p}{A} (T_e - T_i) = F_R (\tau \alpha) I - F_{R U_L} (T_i - T_a) \]  
(5.22)

The measured efficiency is

\[ \frac{m c_p}{A} (T_{\text{em}} - T_i) = (F_R \tau \alpha)_m I - (F_{R U_L})_m (T_i - T_a) \]  
(5.23)

If \( T_i = T_a \), by substituting equation 5.21 into equation 5.23 we find that this equation is equivalent to equation 5.22 and so \((F_R \tau \alpha)_m = F_R \tau \alpha\). So the measured value of \( F_R \tau \alpha \) is the true value, even with a leak at the outlet. So we can rewrite equation 5.23 as

\[ I \frac{F_R \tau \alpha}{A} = \frac{m c_p}{A} (T_{\text{em}} - T_i) - (F_{R U_L})_m (T_i - T_a) \]  
(5.24)

Substituting equations 5.24 and 5.21 into 5.22 gives

\[ (F_{R U_L})_m - F_{R U_L} = \frac{c_p}{A} (m_l) \]  
(5.25)

Note that the difference between the true and measured value of \( F_{R U_L} \) is not dependent on the measured mass flow rates but only on the leak rate.
Figure 5.18 shows the % error of FRUL for different % leak rates both for leaks at the inlet and outlet. Even a leak of only 2% of the mass flow rate at the outlet will cause a 10% error in the measured value of FRUL. Note this analysis only applies for small leaks when the leak is on the outlet as FRα is dependent weakly on plate temperature. If there is a large leak the absorber temperature goes up and so the efficiency decreases.

So even small leaks can have a large effect on the efficiency curve. Also not only is the leak rate important but also the location of the leak. Therefore some measurement of leakage rates need to be carried out. Ideally this would be done with the equipment required for the ASHRAE testing. By blocking off the collector inlet and turning the fan on, the leakage flow rate can be measured through the orifice plate. But the orifice plate is very insensitive to small mass flow rates - Figure 5.19. Likewise, detecting a small pressure drop across the collector with small flow rates is difficult. Cheerna and Mannan [23] have suggested measuring the leakage flow by inserting an extra orifice plate at the inlet. The difference between the two orifice plates gives the leakage flow rate. Given the accuracy of orifice plates, ± 3%, the error in the difference between the two will be greater than 5% of the total mass flow rate, which can give 20% errors in FRUL. Wiles [8] has suggested using a cumulative measuring device, a domestic gas meter with a blocked off collector. Although this enables leaks to be detected, the true leakage
rate cannot be measured as the pressure difference between collector and ambient will be different with the inlet blocked than with it open (see Figure 5.20). Alternatively a simple method for testing for leaks is to measure \( \frac{U_L}{\alpha} \) with no fluid flow and so no leakage flow. This can be done under stagnation conditions if the absorber plate temperature is measured. The Hottel, Whillier and Bliss equation under zero flow is

\[
0 = \tau\alpha I - U_L(T_p - T_a)
\]

(5.26)

So

\[
\frac{U_L}{\tau\alpha} = \frac{I}{T_p - T_a}
\]

(5.27)

This can then be measured at noon under steady state conditions but preferably with reduced insolation, which can be obtained by shading with a screen. This makes the absorber temperature close to that under typical flow conditions and so \( U_L \) will be the same as \( U_L \) under efficiency conditions. \( \frac{U_L}{\tau\alpha} \) measured under stagnation can then be compared with \( \frac{F_R U_L}{F_R(\tau\alpha)} \) measured from the efficiency curve. If these are different then a leak should be suspected.

If a leak is detected, locating the leak can be a problem particularly outdoors where the wind disperses local disturbances of air. One possible method, although requiring rather costly equipment, is to operate the collector at a high temperature, that is with the inlet heater turned up full, and to scan the collector with an infra-red camera.
Air leaks will appear as cold spots at their source.

Air leaks are not much of a problem in air heating collectors when used as part of a domestic heating system, because the air temperature at the inlet is normally close to ambient and the air leaking into the collector is not necessarily at ambient temperature but at house temperature and so this heating of house air can be considered a useful function. Elkin [24] calculated that a 30% leakage of air into a house solar heating system would cause a 33% reduction in its efficiency, whereas such a large leak in a collector during testing would result in an error $\gg 100\%$ in the measured value of $P_{RUL}$.

Analysis of Results and Recommendations for outdoor steady state testing.

To measure the steady state efficiency of the D.C.Hall and S.P. collectors, data were collected during the whole day. The surface of the collector was cleaned at the start of the day. With the fluid inlet temperature set, once pseudo-steady state conditions were obtained the test was started. Tests were performed when the insolation was not up to ASHRAE standard of stability. The minimum insolation level used for tests was 580 Wm$^{-2}$ and measurements were not made before and after noon with four measurements at the same inlet temperature.
For each inlet temperature data were collected for a period greater than the time constant of the collector, and integrated as recommended by the ASHRAE standard. For the D.C. Hall collector the time period was 10 minutes because the measured time constant was 8 minutes, and 5 minutes for the S.P. collector because the time constant was 4 minutes.

Data are read from the data logger tape and stored in the form shown in Table 5.1 by a Hewlet Packard 86 microcomputer. Plots of the data can then be output as graphs on a HP graph plotter, Figure 5.21. The computer also calculates instantaneous efficiency but with no modifications due to angle of incidence. From these data the integrated performance is calculated for each test (Table 5.2a and b) and an efficiency curve plotted using the equivalent normal irradiance, figures 5.22 and 5.23.

A linear regression through the measured points for the D.C.Hall collector, Figure 5.22 gives the following collector performance

\[ \eta = 0.627 - 5.13 \frac{(T_i - T_a)}{I} \]  

(5.28)

and has a correlation coefficient \( r \) of -0.993. The results for steady state testing of the SP collector (Figure 5.23) were taken on some days when the radiation was below 360 Wm\(^{-2}\), so three linear regressions have been plotted giving the following collector performance.
all points \[ \eta = 0.545 - 7.52 \frac{(T_i - T_a)}{I} \quad r = -0.975 \quad (5.29) \]

I>741 Wm\(^{-2}\) \[ \eta = 0.545 - 9.29 \frac{(T_i - T_a)}{I} \quad r = -0.987 \quad (5.30) \]

I<362 Wm\(^{-2}\) \[ \eta = 0.557 - 7.37 \frac{(T_i - T_a)}{I} \quad r = -0.987 \quad (5.31) \]

Equation 5.30 corresponds to the ASHRAE standard curve. The larger insolation produces only a 2% change in the value of \( \eta \), but a 25% change in the value of \( F_{RUL} \). Lower levels of insolation mean lower absorber plate temperatures 30°C for I<362 Wm\(^{-2}\), 50°C for I>741 Wm\(^{-2}\) and so give lower values of \( U_L \). Given that 90% of the solar energy in the UK incident is at levels less than 500 Wm\(^{-2}\), the ASHRAE standard measure value of \( F_{RUL} \) is not representative of collectors operating in the UK.

Top duct collectors like the SP collector have a U-value (and to a lesser extent an \( F_R \) value) particularly sensitive to wind. This is shown in tests 4 and 5 in Table 5.2(b) which shows a drop of 3% in efficiency when the wind speed measured a few metres from the collector (not strictly the meteorological wind speed) increases from 0.4 ms\(^{-1}\) to 2.1 ms\(^{-1}\). For this type of air collector, tests should thus be carried out under more defined wind conditions than laid down by ASHRAE (<4.5 ms\(^{-1}\)).

Tully and Wernick [25], have investigated the variation in efficiency which can result from a 0-4 ms\(^{-1}\) variation in wind
speed, within the ASHRAE standard for a water collector (see Figure 5.24). They suggest the use of the normalising function to produce a unique efficiency curve, Figure 5.25. But the effectiveness of such normalising functions has not been proved because their work is based on computer simulations and has not been validated under real weather conditions. In particular the simulation uses the wind heat transfer coefficient proposed by McAdam in 1954 [26].

\[ h_w = 5.7 + 3.8V \]  \hspace{1cm} (5.32)

The validity of this equation is questioned in Section 4.2.1 because no account is taken of the angle of tilt of the collector, or its geometry and surface properties which influence the wind coefficient. Also the use of such a normalising function would be different for different types of collector, and in the case of air heating collectors there are almost as many collector types as collectors in operation.

The efficiency of air collectors is dependent critically on the mass flow rate [27]. It is therefore important to lay down some guide lines for the variation of mass flow rate during a test, and from test to test. This can be difficult because the mass flow rate from the fan changes with temperature. It is not clear whether this change in flow rate should be allowed during testing because under real operating conditions changes in flow rate will occur - or whether a constant flow rate should be maintained for testing.
During the experimentation the mass flow rate was seen to change more on windy days than on windless days, see Figure 5.26. Therefore if an open loop circuit is used the fluid inlet and outlet should be sheltered from the wind.

Conclusions.

The ASHRAE standard can be used for testing air heating collectors outdoors in an open loop circuit with relatively little expense. However, if tests are to be carried out under the limited environmental conditions suggested by the standard then testing within the UK is limited to a few days a year, and would typically require a year to establish an efficiency curve, making the method very expensive in time. Furthermore, the International Energy Agency has reported [28] a large variation in results between one ASHRAE type test site and another, because of instrumental errors and climatic changes within the standard limits: See Figure 5.27. These results are for liquid collectors. For air collectors the spread is expected to be even larger, particularly when testing top duct collectors, which are very sensitive to the effect of wind. It is suggested that such collectors should only be tested under specified wind conditions. To achieve this, artificial ventilation of the front cover may be necessary. Air collectors are also very sensitive to changes in mass flow rate, so limits should be placed on the fluctuation of flow rate during the tests.
Also, air leaks into the test equipment and collector can alter the measured efficiency. The effect a leak has is not only dependent on its magnitude but also its location. A measurement of the leakage rate should therefore be made, and if such equipment is not available then a method of testing under stagnation is suggested. To reduce the time period required it is suggested that several measurements at different fluid inlet temperatures be made during the same day on clear days and that adjustments due to the diurnal transient of the effective heat capacity be adjusted as suggested by Aranovitch [5] (see Equation 5.4). But this method is only recommended for collectors with time constants <5 minutes. Air collectors can have long time constants under flow conditions, so steady state testing on a fixed rig is obviated by the diurnal variation. It is therefore recommended that long time constant collectors are tested using tracking mounts [29]. Long time constant collectors also require stricter control over inlet fluid temperature to monitor accurate values of energy collected, as suggested by ASHRAE [29].

Testing collectors in the UK under steady-state conditions is very difficult because of the diffuse and transient nature of the insolation which makes results unrepresentative of typical operating conditions of solar collectors. It is therefore proposed that transient tests should be used. These are easier to perform and are more representative of collector operation in the UK.
5.3.2. **Transient Testing**

Under steady state testing, the rates at which energy falls as radiation on the collector and re-emerges, either as heat loss or in the heat transfer fluid, are constant. Thus the balance between these energy flows may be expressed by the Hottel-Whillier and Bliss equation. This takes no account of the rate at which energy is transferred within the collector. To test in unsteady conditions we must use an equation which accounts for the heat flow into and out of the components of the collector as their relative temperatures change. For this 'transient' model to be applicable to all collectors, the process must be described in such a way as to cope with all possible weather fluctuations and without reference to the specific construction and material properties of the collector.

The test method is substantially the same as for outdoor steady-state tests, except that integrated values over a number of time intervals are taken for each of the variables, and the range and variability of the irradiance and ambient temperature are unrestricted. The characteristic steady-state efficiency curve for the collector is recovered from the transient data by assuming a model which is a time-dependent generalisation of the Hottel-Whillier and Bliss equation. Such a method for transient testing has been drafted for a British Standard DD77:1982 [30]. A programme
of experimental validation has shown that transient tests agree with steady state tests to within 3% for a range of liquid flat plate collectors. Although the theory of transient testing is not limited to liquid collectors, the conditions specified in the British Standard are with liquid collectors in mind, and no transient testing of air collectors has been reported. This section examines the alterations needed to the Draft British Standard to make it applicable to air collectors, and reports the results of testing two-types of collector (D.C.Hall and SP - see section 5.1.1).

Theory of Transient Testing

Although the method of transient testing is less restrictive than steady state, the theory and data analysis are far more complicated, and require the use of a computer.

The simplest transient model of a flat plate solar collector is the model of Klein et al [31] see Section 4.2.2, which assumes the entire collector can be treated as a unit, insofar as capacitance effects are concerned, so the capacitance \((mc)_e\) is assigned a single node, and the same temperature distribution across the absorber as the heat transfer fluid. From this Taylor [32] proposed as a method of analysing data.
\[
\frac{mc_p}{A} \left[ T_e(t) - T_i(t) \right]
\]

\[
= \int_0^\infty F_R \kappa(\tau) \left[ I(t-\tau)(\tau a)_c - U_L \left( T_i(t-\tau) - T_a(t-\tau) \right) \right] d\tau \quad (5.33)
\]

where \( \kappa(\tau) \) is a linear response function of unspecified form but having a cut off time \( \tau_d \), earlier than which the "memory" of the collector is negligible i.e. \( \kappa(\tau) = 0 \) at \( \tau > \tau_d \). Both \( U_L, F_R \) and \( m \) are assumed to be independent of \( t \), but irradiance \( I \), and ambient temperature \( T_a \) are allowed to vary with time.

For consistancy with the steady-state equation it is necessary for \( \kappa(\tau) \) to satisfy the normalisation condition

\[
\int_0^\infty \kappa(\tau) d\tau = \int_0^{\tau_d} \kappa(\tau) d\tau = 1 \quad (5.34)
\]

If \( T_i \) is constant then \( \Delta T(= T_i - T_a) \) is slowly varying and therefore we may assume that the output power per unit aperture area conveyed by the heat transfer fluid is

\[
q(t) = \frac{mc_p}{A} \left[ T_e(t) - T_i \right] = F_R(\tau a)_c
\]

\[
\int_0^{\tau_d} \kappa(\tau) I(t - \tau) d\tau - \frac{F_R U_L}{\tau_d} \int_0^{\tau_d} \Delta T(t - \tau) d\tau \quad (5.35)
\]

For data analysis the time is made discrete, and thus \( \tau_d \) is broken into \( N \) increments. Thus,
\[ q(j) = F_R(\tau\alpha) \sum_{n=1}^{N} k_n I_n(j) - F_R U_L \bar{\Delta T}(j) \quad j=1,2,3,...J \quad (5.36) \]

where \[ \bar{\Delta T}(j) = \frac{1}{N} \sum_{n=1}^{N} \Delta T_n(j) \quad (5.37) \]

and \( I_1(j) \) and \( \Delta T_1(j) \) are values integrated over the same time increment as \( Q(j) \), but \( I_2(j) \) and \( \Delta T_2(j) \) are values integrated over the previous time increment to \( Q(j) \) up to \( N \) successive previous time increments. It is upon equation 5.36 that the transient test is based.

With \( N \) fixed equations 5.36 is linear in the \((N+1)\) unknown parameters \( F_R U_L, F_R(\tau\alpha) \cdot k_1, F_R(\tau\alpha) \cdot k_2, \ldots, F_R(\tau\alpha) \cdot k_N \). If we make \( J \) measurements where \( J = N+1 \), we will have \( N+1 \) equations which can be solved to obtain the \( N+1 \) unknown parameters. But with \( J > N+1 \) some criterion of best fit is required for the unknown parameters since there will normally be errors of measurement in the known parameters, \( q(j), I_1(j), \ldots, I_N(j), \Delta T_1(j), \ldots, \Delta T_N(j), j = 1 \) to \( J \).

Rogers [33] proposed the use of an 'instrumental variable' instead of the standard least squares fit to arrive at a best fit for the unknown parameters. This is because statistical analysis has shown that the least squares fit gives biased estimates for the unknown parameters when there are errors of measurement in all the variables. Once the \((N+1)\) unknown parameters have been calculated with the use of an 'instrumental variable' the value of \( F_R(\tau\alpha) \) can be deduced.
using the normality condition

\[ \sum_{n=1}^{N} P_{R}(\tau \alpha) \cdot k_n = P_{R}(\tau \alpha) \cdot \sum_{n=1}^{N} k_n = P_{R}(\tau \alpha). \]  

(5.38)

and so \( k_1 \ldots k_N \) can be found and used to generate two new variables independently of the estimated value of \( P_{R}(\tau \alpha) \) and \( P_{RU_L} \)

\[ \eta(j) = \frac{q(j)}{\sum_{n=1}^{N} k_n I_n(j)} \]  

(5.39)

and

\[ T^{\ast}(j) = \frac{\Delta T(j)}{\sum_{n=1}^{N} k_n I_n(j)} \]  

(5.40)

If the assumed model is correct, it follows that these variables lie on the steady state efficiency plot

\[ \eta(j) = P_{R}(\tau \alpha) \cdot P_{RU_L} T^{\ast}(j) \]  

(5.41)

where \( T^{\ast}(j) \) is the reduced temperature \((T_i - T_a)/I\). Plotting \( \eta(j) \) against \( T^{\ast}(j) \) and carrying out a least squares fit enables \( P_{R}(\tau \alpha) \) and \( P_{RU_L} \) to be calculated. The closeness of these values to those calculated from the values obtained directly from the 'instrumental variable' is a measure of the validity of the model. The accuracy of the test procedure can be assessed by comparing the estimated efficiency
parameters $F_R(\tau a)^*$ and $F_{RUL}$ with those obtained by the steady-state test.

The *British Standard DD77:1983* is based on the above approach. It states the conditions for successful transient testing of liquid collectors. They differ from steady state testing in that the three variables, $\Delta T$, $q$ and $I$ are required as integrated values over successive time increments equal to one fifth of the cut-off time $\tau_d$ of the response function. For liquid collectors $\tau_d$ is approximately the discharge time over which a "thermal mass" ($mc$) of fluid equal to the thermal mass of the collector (including the fluid) passes through the collector. When the flow rate is constant,

$$\tau_d = \frac{(mc)e}{m c_p} \quad (5.42)$$

Since for a liquid collector most of the effective mass is in the fluid the discharge time is approximately the time taken for the fluid to pass through the collector. Thus,

$$\tau_d \approx \frac{M}{m} \quad (5.43)$$

Where $M$ is the fluid capacity of the collector. The time increment for data integration is then

$$\frac{M}{5m} \quad (5.44)$$

The equivalent normal solar irradiance should be calculated
from direct and diffuse measured irradiance for every time increment from equation 5.14. This is important because higher diffuse fractions will be expected during the transient testing than steady state testing, effecting the value of \((\tau_0)\). See Figure 5.28 [34]. Thus transient testing requires an extra pyrometer to measure the diffuse irradiance continuously. Potentially the most serious error in transient testing, as in steady state testing, is in the measurement of solar radiation. Because the errors are relatively greater for small values of irradiance the standard specifies that not more than 20\% of the values of total solar irradiance in the plane of the collector shall be less than 300 \(\text{Wm}^{-2}\). The importance of accurately measuring the direct and diffuse irradiation has shown errors of the order of 10\% in estimating \(F_{RUL}[33]\).

The fluctuations in the fluid-inlet temperature shall not exceed \(\pm 0.2^\circ\text{C}\) throughout each data sequence, because a drift of more than \(0.1^\circ\text{C}\) in one hour with a high inlet temperature will result in an error greater than 1\% of the \(F_{RUL}[33]\). The flow rate shall not vary by more than \(\pm 1\%\) because a constant drift of \(\epsilon\%\) of the thermal mass in the heat transfer fluid will give roughly the same percentage error in the estimates of the collector parameters.

The data are to be collected for at least 4 different values of inlet temperatures, covering the whole range of possible inlet temperatures. A period of at least 20 time increments
with fluid flow for preconditioning of the collector must be allowed at each fluid inlet temperature before data are collected over a further period of at least 60 consecutive time increments, which is approximately equal to \(12\tau_d\) for liquid collectors.

The data are then analysed by a 'Fortran' programme listed in Appendix E of the British Standard. The three variables \(I, q\) are \((T_i - T_a)\) are input as a series of averages for consecutive time increments. A serial number shall be associated with each set of values of the three variables, such that consecutive serial numbers are associated only with consecutively measured values. A trial value of 5 for \(N\) is input into the program which then calculates the values of \(F_R(\tau\alpha)\cdot k_1, F_R(\tau\alpha)\cdot k_2 \ldots F_R(\tau\alpha)\cdot k_n\) and \(F_{U_L}\), along with their estimated error. The variables \(\eta(j)\) and \(T^*(j)\) of equations 5.39 and 5.40 are plotted. A least squares fit each way of this plot is then obtained to check the consistency of the thermal performance characteristic. The consistency is established if the calculated values of \(F_R(\tau\alpha)\cdot\) and \(F_{U_L}\) lie between the values of each obtained from the least squares fit. The value of \(N\) is then increased until the minimum value of estimated standard error in \(F_{U_L}\) is obtained.

The British Standard makes the outdoor transient measurements compatible with the British Standard for indoor steady state testing [30] (which is similar to the American Standard) the results of which are presented in the form
\[ \eta = F_{ave} (\alpha T) - U_L F_{ave} \frac{(T_m - T_a)}{I} \quad (5.45) \]

This is with respect to the mean fluid temperature \( T_m \), which is the average temperature of the heat transfer fluid in the collector. Its value is taken to be the arithmetic mean of the fluid inlet and outlet temperatures: \( F_{ave} \) is approximately the same as \( F' \), because \( F' \) applies to the true average fluid temperature [9]. So equation 5.45 becomes

\[ \eta = F'(\alpha T) - U_L F' \frac{(T_m - T_a)}{I} \quad (5.46) \]

Dividing equation 5.41 by \( F'' \) gives

\[ \frac{\eta(j)}{F''} = \frac{F}{F''} (\tau \alpha)_o - \frac{F}{F''} U_L T^*(j) \quad (5.47) \]

Substituting from Duffie and Beckman [9]

\[ \frac{F}{F''} = F' \quad (5.48) \]

we get

\[ \frac{\eta(j)}{F''} = F'(\tau \alpha)_o - F'U_L T^*(j) \quad (5.49) \]

and \( F'' \) can be obtained from substituting equation 5.48 into 4.18 and rearranging to give

\[ F'' = \frac{\frac{F' U_L A}{mc_p}}{\ln \left(1 - \frac{F' U_L A}{mc_p} \right)} \quad (5.50) \]
Thus $F'$ can be calculated by substituting the calculated value of $F_{RU_L}$ into equation 5.50 and then $F'(\alpha \tau)_o$ and $F'UL$ can be calculated from equation (5.49) by plotting $\eta(j)/F''$ against $T^*_o(j)$. Hence we get $F_{ave}(\alpha \tau)_o$ and $F_{ave}UL$, because they are approximately the same as $F'(\alpha \tau)_o$ and $F'UL$.

**Data Analysis**

For consistency a version 'TRANS' (see Appendix D) of the British Standard computer programme was written in Basic, outputting data in the format specified in Appendix F of DD77:1982. 'TRANS' was written to run on a 'Hewlet-Packard 86' microcomputer. Care should be taken in running the program on other computers because the original Fortran version is written in double precision (14 digit precision) and Rogers has reported problems of instability in the program when run in single precision [35] (7 digit precision). This is due to the large number of matrix manipulations in the program. The Hewlet Packard works to 12 digit precision and no difficulties were experienced in running the program.

'TRANS' was checked in three ways. First steady-state data were fed in and the program was run with $N=1$, and the output examined to see if the same answer was obtained as for outdoor conventional steady state analysis. Second, perfect one node dummy data were fed into the computer and the values of $k$ checked with the original values used to generate the data. Third, the program was tested to see if in theory it
could operate with data from an air heating collector (see next sub-section). The multi nodal computer model RRDCT was used to generate data for both a light weight and a heavy weight collector for transient weather conditions and for steady state conditions, for four inlet temperatures. The transient results analysed by TRANS were then compared with the steady state results. This proved to be a very useful exercise, and would provide an ideal tool for investigating the effects on the transient test procedure of small changes in input temperature, mass flow rate or ambient temperature.

**Transient testing of air collectors**

So far we have considered the transient testing procedure mainly in relation to liquid collectors. There are several important differences between air and liquid collectors which produce extra problems when transient testing air collectors. Most of the effective thermal mass in liquid collectors is in the fluid, whereas for air collectors the collector materials form most of the effective heat capacity. Also the fluid flow rate \( m \) is much faster in air collectors than in liquid collectors and the heat transfer from the absorber to the fluid is less in air collectors. This means that collector materials and type of fluid flow (turbulent or laminar) become more important in the effective heat capacity, whereas for liquid collectors only the fluid flow rate \( \text{rate} \) will be dominant. So, for air collectors the definition of cut off time \( \tau_d \) expressed in equation 5.43 is not expected to hold, instead \( \tau_d \) will depend on the effective heat capacity, \( (mc)_e \)
which in turn can be represented by the time constant of the collector, which can be measured by screening the collector—see Section 5.3.1. Although this method requires steady state conditions they need to last for only the order of one time constant. However, it is not essential to know the exact value of $\tau_d$—an approximate value will do so that the initial magnitude of the time increments $\Delta t$ in data recording can be calculated from $\Delta t \propto \text{time constant}$, $N$ can then be increased until the optimum is found which is when the standard error of $F_{RUL}$ is at a minimum.

Air collectors can have a smaller effective heat capacity than liquid collectors by careful design as the fluid has a small heat capacity and the collector materials can be lighter for an air collector, because the weight of the fluid is less. But in general air collector time constants under flow conditions are longer, 3 to 15 minutes compared to those of liquid collectors, 1 to 5 minutes. This is because the air usually extracts heat from the collector at a lower rate than the liquid in a liquid heating collector.

Maintaining a constant inlet temperature, which is one of the conditions for transient testing, is more difficult with air systems, particularly if testing is under open circuit conditions, where the inlet air temperature is governed by the fluctuations in ambient temperature. A closed system with a regenerator would enable more stability of the inlet temperature but would increase the cost of any such test system. The importance of inlet temperature variation for
air collectors is not known and requires more investigation. The variation of $T_i$ effects both $\dot{q}$ and $T^*$. The effect on $\dot{q}$ is anticipated to be the greater, but less crucial in air collectors due to most of the heat capacity being in the structure of the collector and not within the fluid.

The model used for transient testing is only strictly true under conditions of constant mass flow rate, more so for air collectors whose efficiency is more dependent on mass flow rate, than is the case for liquid collectors. Constant mass flow rate is difficult to achieve for air collectors where $m$ is a weak function of $T_e$ if the fan is at the outlet, and of $T_i$ if the fan is at the inlet. This is particularly important for low effective mass collectors where the exit fluid temperature tends to fluctuate along with the insolation. Thus, under conditions of fast transients the mass flow rate will change rapidly. If the volumetric flow rate is to be measured, care must be taken for correcting the density dependence on temperature, because the flow rate needs to be measured at the exit. Adjustments for the temperature dependence of the specific heat capacity are not considered necessary because $c_p$ (air) only changes by 0.6% over a 100°C temperature difference. This introduces an insignificant error in comparison with that from the mass-flow rate, which is also used to calculate $\dot{q}$, unless more accurate methods of flow measurement are developed or collector testing is required over a greater range of temperatures.
The validity of transient testing for air collectors with different masses was tested using a multi nodal model of a rear duct collector, RRDCT (Appendix C).

Steady state efficiency curves were obtained for two typical air-heating collectors differing only in their mass, by running RRDCT under ASHRAE test conditions - see Figure 5.29. The collector properties are shown in Table 5.3. The two collectors only differed in the plate thickness of the absorber and rear duct. One collector absorber and rear duct was "made" from 0.5mm thick aluminium plate and so would have a similar mass to that of the SP collector while the other collector's absorber and rear duct were of 2mm thick aluminium, and so had a similar mass to the D.C.Hall collector. The steady state curve was not affected by mass.

To obtain a transient pseudo steady state curve, RRDCT was run under transient diffuse radiation. Figure 5.30. The response of both the collectors with a fluid inlet temperature of 70°C is shown in Figure 5.31. The data were generated at 1 minute intervals for the 0.5mm collector and every 4 minutes for the 2mm plate collector. Figure 5.32 shows the integrated response of the two collectors. 30 minutes of data were generated for the 0.5mm collector and 120 minutes for the 2mm collector giving 30 data points for each collector. Only 30 data points were used, rather than the 60 recommended by the B.S. draft, in order to reduce computing time, however the extra errors that this created were established and shown to be small.
The collector's transient responses were also calculated for fluid inlet temperatures of 12°C, 40°C and 70°C. These data were then analysed by TRANS (Appendix D). Figure 5.33 shows the predicted values of $F_R U_L$ and $F_R(\tau\alpha)_o$ along with their standard errors, for different values of $N$. For the 0.5mm plate collector the smallest value of $e F_R U_L$ corresponds to $N=6$ and for the 2mm plate collector $N=5$. The two response functions are plotted in Figure 5.34. This suggests a cut off time $\tau_d$ of 6 minutes for the 0.5mm collector and 20 minutes for the 2mm collector. If we compare $\tau_d$ to the collector time constant $\tau_c$ which has been calculated by running RRDCT under ASHRAE collector time constant conditions (Section 5.3.1) (see Figure 5.35) we see that $\tau_d = 2\tau_c$. This is not the case with the liquid collectors, whose time constants under flow are of the order of 1 minute with cut off times of the order of 5 to 12 minutes [36].

Figure 5.29 shows the data points plotted for the 0.5mm plate collector, for $N=1$, that is with no correction for what has happened in the past to the collector, and for $N=6$ when the data are corrected by the response function. Those data lie on a pseudo steady state curve that is quite different from that produced by the ASHRAE test. Table 5.4 shows the values of $F_R U_L$ and $F_R(\tau\alpha)_o$ corresponding to the smallest error, along with the steady state values. The first thing to notice is the big difference between the values of $F_R\tau\alpha$ for steady state and transient. This is due principally to the difference in the transmissivity of the cover to direct
normal and diffuse radiation. This highlights the importance in correcting for this factor by measuring the direct and diffuse components and using an angle of incidence modifier. The corrected values are also shown in Table 5.4 as \( K_F R_T \alpha = F_R(\tau \alpha) \). The biggest discrepancy now is between the values of \( F_R U_L \) for steady state and transient. This is due to the lower average plate temperature (40°C less) and the higher sky temperature experienced under transient conditions. According to steady state theory these two effects reduce the \( U \) value by \( \approx 0.3 \) and increase \( F_R \) by \( \approx 0.01 \). The corrected steady state values then become \( F_R U_L = 2.55 \) and \( F_R(\tau \alpha) = 0.684 \). Given these adjustments, there is good agreement between the corrected values from the steady state model and the transient model for the 2mm plate. But the agreement for the 0.5mm plate is poor. Both the values of \( F_R \tau \alpha \) and \( F_R U_L \) are low for 0.5 mm plate. These discrepancies cannot be explained by any temperature effect or mathematical averaging because both the 0.5 mm and 2 mm collectors have the same average absorber temperature and both have the same number of data point. Moreover roughly the same values of \( N \) were used therefore the same values of \( F_R(\tau \alpha) \) and \( F_R U_L \) are expected for the 0.5 mm plate as for the 2 mm plate. The discrepancy must therefore be due to the difference in mass, and may be due to the 0.5 mm plate having a response time similar to the period of variation of the solar radiation.

To summarise, there appear to be no theoretical reasons why
the transient technique cannot be adapted for air collector testing. The pseudo steady state curve generated under transient conditions will be slightly different for transient testing than ASHRAE testing as $P_R$ will be higher and $U_L$ lower under transient conditions due to different plate and sky temperatures. But corrections must be made for the difference in transmissivity of the cover under diffuse and direct conditions. The data should be sampled every $2T_c/5$, which for most air collectors will be from 1 min to 5 min. Data should not be sampled every $M_{05}$ as is the case with liquid collectors as the majority of the heat capacity is not in the fluid but in the collector structure. Care should be taken when using data collected over periods when the radiation varies with periods equivalent to the collector response time, which is approximately twice the collector time constant.

Outdoor Transient Testing Results

To substantiate the above results of numerical testing the D.C.Hall and SP collectors were tested outdoors under transient conditions and the results compared with those obtained while testing the collectors under steady state conditions. The differences between the two collector types top and rear duct, and small and large effective heat capacity, provide a useful test for the application of this method to air collectors.
The fluid capacity of the collector $M$ is 0.019 Kg, with a mass flow rate of 70 Kg hr\(^{-1}\). The fluid transit time is therefore $\frac{M}{m} \approx 1$ second. The time taken for a thermal mass of fluid equal to the thermal mass of the collector to pass through the collector, the thermal mass transit time, is given by

$$\frac{(mc)_e}{mc_p}$$

from section 5.3 is 3,300 J°C\(^{-1}\)

So far a 70 Kg hr\(^{-1}\) mass flow rate the thermal mass transit time is \approx 3 minutes. A time constant of \approx 4 minutes was measured for the collector. It was therefore decided to base the scan period for data collection on $2\tau_C$ rather than of $(mc)_e/mc_p$. A scan period of 1 minute was chosen.

Table 5.5 shows the data collected on the three days with four inlet temperatures. On one of the inlet runs 60% of the data were below 300 Wm\(^{-2}\). Only one solarimeter was available and so accurate corrections for the relative proportion of diffuse and direct radiation for each measurement was not possible. However, a correction factor was used as in the steady state analysis, see Equation 5.16. The data were analysed using TRANS for $N = 1$ to 8 (larger values of $N$ would have involved unacceptably long programming times). Table 5.6 shows the output for $N = 1$ of TRANS in the format specified by DD77:1982, Appendix F.6.1. For $N = 1$, $k_1 = 1$ and so no
adjustment is made for what has happened to the collector over previous time increments. Therefore the analysis is similar to that for the steady state, i.e. the previous conditions are assumed the same as the present. The data are plotted in Figure 5.36. Because the insolation was transient the previous insolation was not the same as the present insolation (see Figure 5.37), and so the scatter in the points is larger than when the insolation was steadier. The dashed lines are the maximum and minimum least squares fit of the data, and the dashed line is from the values of $P' (\tau \sigma)$ and $P'UL$ obtained directly from the 'instrumental variable'. This line does not lie between the two least squares fit, showing that the model is not applicable for $N = 1$.

For $N = 1$ the percentage standard error in $FRUL$ is 15%. The percentage standard error in $FRUL$ versus $N$, the number of post time intervals used in analysing the present collector status, is shown in Figure 5.38. The minimum value of $\hat{\sigma} FRUL$ occurs when $N = 7$. Table 5.7 shows the data output for $N=7$ and figure 5.39 shows the data plotted. Comparing the data for $N = 1$ and $N= 7$ in figures 5.36 and 5.39 the spread of data is less for $N=7$. This is shown in the closeness of fit in the maximum and minimum least squares fit, and in the estimated efficiency curve lying between the two lines of best fit. So the
transient model is consistent for \( N = 7 \). Which suggests that the cut off time \( (\tau_d) \) of the collector is 7 minutes, approximately twice the collector time constant. Figure 5.40 shows how the response function \( k_n \) varies with \( n \). As expected the present level of insolation is the most important in determining the collector's present performance, with previous levels of insolation having a progressively smaller effect.

The steady state efficiency curve for this collector derived from transient testing, modified for angle of incidence is given by

\[
\eta = 0.520 - 8.48 \frac{(T_i - T_a)}{I}
\]  

(5.51)

With a 5% standard error in \( F_{RUL} \) and a 2% standard error in \( F_R(\alpha) \).

If these results are compared to the results in equation 5.30 obtained for ASHRAE steady state testing then the values of both \( F_R(\alpha) \) and \( F_{RUL} \) are seen to be lower under transient testing than in the steady state, although perhaps not significantly if the ± 6% possible error reported by Aranovitch [22] is taken into account. The lower value of \( F_{RUL} \) under transient conditions can be explained by the lower plate temperature arising from lower insolation.
levels under transient testing. This same effect was observed when steady state tests were carried out at lower insolation levels (see equations 5.31). The lower $F_{R}(\tau \sigma)$ can be explained by an increase in the diffuse fraction of radiation under transient testing which was not fully accounted for because only one solarimeter was available, therefore the diffuse fraction was only measured after the experiment, and so only one correction value could be used to convert from the measured total solar irradiance to the equivalent normal solar irradiance. The rather large error in $F_{RUL}$ could be reduced if more than 20 data points ($J = 20$) had been recorded for each inlet temperature, and had each set of readings been taken under similar wind conditions (see Figure 5.37 for wind conditions during transient testing).

D.C.Hall Collector

The thermal time constant of the D.C.Hall collector is $\approx 7$ minutes (air flow rate $= 65$ kg hr$^{-1}$) and thus the cut off time $\tau_d$ for the response function is expected to be $\approx 14$ minutes. Data were recorded at one minute intervals and then averaged over two minute intervals by the computer. Thus $\frac{2I_c}{5} = \Delta T$.

Five hours of data were collected with four different fluid inlet temperatures, and then analysed by TRANS
with different values of N. The best fit was for N = 7, see Figure 5.41. This agrees with the cut off
time of 14 minutes.

The transient efficiency corrected for angle of
incidence is

\[ \eta = 0.583 - 5.19 \frac{(T_i - T_a)}{I} \]  

(5.52)

with a 5% standard error in $F_{RUL}$ and 2% standard
error in $F_R(\tau\alpha)$. Again if one compares these
results with those obtained under steady state
testing (equation 5.28), the value of $F_R(\tau\alpha)$ is
lower, because there is no correction for the
increased diffuse fraction was not sufficient, but
the value of $F_{RUL}$ is higher and not lower than
expected. This can be attributed to the higher wind
speed 5.2 ms\(^{-1}\) in the transient case, compared to an
average of 2 ms\(^{-1}\) in the steady state. Again the
same order of error was obtained as with transient
testing of the SP collector.
Conclusions

The method of transient testing can produce results comparable to steady state testing, and is particularly useful in the varying climatic conditions of the UK, where a collector can now be tested in a single day. ASHRAE steady state testing conditions in the UK prevail for less than 168 hours a year ($I > 600 \text{ Wm}^{-2}$, steady, and $\theta < 30^\circ$) whereas conditions suitable for transient testing ($I > 400 \text{ Wm}^{-2}$ and $\theta < 60^\circ$) occur for 1056 hours, permitting the transient testing of collectors even during the winter months [36]. Although the numerical computation for transient testing is more complex than steady-state, micro computers are now available which can handle such complex tasks, and with the compatibility of data loggers and micro-computers the data handling can be carried out easily and cheaply. However, transient testing requires an extra pyrometer to adjust for the relative magnitude of direct and diffuse radiation. Improvements in pyrometer accuracy at low intensities would allow the testing of collectors under even more varied climatic conditions.

The efficiency curve obtained during transient testing is different from that obtained under steady state conditions due to the different climatic conditions. Thus, lower insolation levels cause lower average plate temperatures, more diffuse radiation reduces transmission, and transient conditions normally go with windy conditions so increasing
the heat loss.

The transient testing of air collectors presents no theoretical problems, but more detailed analysis is required to determine the precision with which the testing conditions have to be specified. In particular, the effects of variations in wind speed, mass flow rate and fluid inlet temperature have to be investigated and a multi-nodal computer model provides a useful tool in such studies. The sampling time interval for air collectors should be $\frac{2\tau_c}{5}$, where $\tau_c$ is the thermal time constant under flow conditions.
5.4 Indoor Testing

The difficulty in obtaining steady state conditions outdoors can be overcome by indoor testing. Guidelines for indoor steady state testing are laid down by ASHRAE [2] and by the British Standards Institute [6]. The major problem with indoor testing is trying to simulate the solar spectrum, solar angle of incidence and effective sky temperature. Usually either a xenon lamp [37] or a compact source iodine (CSI) lamp [38] are used to provide the solar radiation. The only solar test system based in the United Kingdom is at University College Cardiff [39], and uses CSI lamps. The system cost £20,000, and was funded by SERC. However, the CSI lamps do not fulfil the ASHRAE requirements for spectral quality or collimation. The irradiance can vary randomly by as much as ± 10% and the lamps, output and spectrum change with age. Nevertheless, outdoor steady state results agree to within ± 5%, the discrepancy being accounted for by the lower beam collimation and the higher sky temperature experienced in the simulator. Ideally a test facility similar to that at Cardiff should be used to standardise collectors. However, such facilities may prove too costly to carry out research and development of collector designs. This is particularly the case for air collectors where a long test time is required to fully characterise an air collector because of its strong dependence on air mass flow rate. Facilities larger than those presently used at Cardiff may also be required to test air collectors. The maximum
collector length which can be tested at Cardiff is 2.24 meters, whereas air collectors are often designed to span the whole length of a roof.

A small indoor test facility was built at The Open University in order to verify computer models and to examine the feasibility of cheap indoor testing facilities for air collectors. Figure 5.42 shows a schematic diagram of the test rig. The monitoring equipment and fluid flow into and out of the collector are identical to the outdoor test facility (see Section 5.3). The radiation source consists of 16, Thorn PAR 38 tungsten filament sealed beam reflector 'Cool Ray' lamps E27 (ES), 150W at 240V. These produce a high colour temperature (4000 K) whilst maintaining a long life time (>1000 hrs), through the use of a dichroic filter which transmits much of the infrared radiation through the rear of the bulb. The surface behind the lights was painted black to absorb this radiation. The relative spectral intensity of the lights was measured with a grating monochromator and Golay detector cell (corrections were made for the grating blazing function). Figure 5.43 shows the spectral intensity of a 'Cool Ray' light along with the solar spectrum for air mass 2.

The 'Cool Ray' lamps produce more infrared radiation than does the Sun. For this reason selective absorbers which have a cut off at 1.8μm (see Figure 5.43) should not be used. Absorbers with absorbtivity independent of wavelength pose no
problems. The spectral transmissivity of 2mm polycarbonate sheet is also shown in Figure 5.43. The total transmissivity of 2mm polycarbonate to 'Cool Ray' radiation was measured by covering a Kipp and Zonan with a sheet of polycarbonate. The transmissivity was 79% compared to 86% for solar radiation. The difference is due to the spectral distribution and that in the test rig 89% of the radiation was diffuse, as measured in accordance with the ASHRAE standard.

The distance between the lights and the collector surface was 1.6 metres. The lamps were arranged in a hexagonal pattern with the sides of the lighting rig enclosed in aluminium foil screens because this was found to substantially increase the intensity of radiation and to improve the intensity distribution across the collector. The total illuminated area was 2.7 m². The uniformity of radiation across the plain of the collector was measured with a Kipp and Zonan solarimeter at 16 spots across the collector surface. The solarimeter was allowed to stabilise for 1 minute before data were integrated at each location for 1 minute. Figure 5.44 shows the intensity distribution across the collector plane. The maximum deviation from the average ($= 200 \text{ Wm}^{-2}$) is $\pm 9\%$ which compares well with results at Cardiff of $\pm 20\%$ [39]. Large non-uniformity for most collectors should not present much of a problem as most solar collectors act as good integrators. However for plastic air collectors such as the SP collector (Section 5.1.1) this may prove to be a problem, because plastic is a bad conductor. If local heating occurs
say near the air inlet then the absorber being an insulator will mean there will be a different temperature gradient across the collector than had the local heating been at the output. The intensity distribution was originally measured across the collector before and after each test. However, this was found not to be necessary provided that the lamps had been allowed to stabilise for one hour before the test. The radiation was continuously monitored at the side of collector during testing to detect the failure of a lamp.

The effective sky temperature was not measured, because the equipment was not available. Nevertheless it can be assumed to have been higher than ambient, the opposite of the usual outdoor case.

The laboratory containing the test rig was air conditioned and the temperature could be controlled to ± 1°C. However, the ambient temperature was found both difficult to control and difficult to measure. This was due to the screening around the solar simulator and the small size of the room. The British Standard draft DD77:1982 recommends that the ambient temperature be measured at the outlet of the wind generator. However, the air coming from the wind generator was approximately 1°C above ambient, and although this is within the specification of BSI DD77:1982, this temperature difference could have an appreciable effect on the results because of the low intensity radiation source, and the consequent small rise in air temperature across the
collector. The temperature of the air coming from the wind generator was used in the following analysis for the ambient temperature because it was felt to be the most representative value of ambient temperature available.

The wind was simulated by a centrifugal fan connected to an aluminium manifold - see Figure 5.45. This could provide an average wind speed of 0 - 5 ms\(^{-1}\) across the collector surface. Wind speed was monitored at 16 points across the collector using a hot wire anemometer. The centrifugal fan was however not considered to be a very good simulator of natural wind conditions because the distribution of wind speeds across the plane of the collector varied by a factor of two (Figure 5.46), and because the generator destroyed any boundary layer which would have been present in outdoor testing, creating an artificially high wind speed indoors.

ASHRAE does not lay down any recommendations for wind speed during indoor testing and the British Standard Draft recommends a mean wind speed of 2 - 8 ms\(^{-1}\) at a height of between 25 mm and 50 mm above the collector. Green et al [40], who have investigated the relationship between heat loss due to wind for indoor testing and outdoor testing, have taken measurements of wind speed at 75mm above the collector cover there is therefore a need to standardise the measurement height of wind speed. There also seems to be some confusion as to how to relate the air velocity parallel to a collector as measured in solar simulators to the meteorological wind speed measured in outdoor tests and
normally used in the empirical correlations used in computer modelling (see Section 4.2.1).

Oliphant [41] has shown that measured meteorological wind speed can be between 1.3 and 3 times greater than the air velocity parallel to the collector, depending on the wind direction, and Green et al [40] have shown that a 1 ms⁻¹ air velocity parallel to the plane of the collector measured a distance of 75 mm above the collector gives a measured wind heat loss comparable to that predicted by a 3 ms⁻¹ meteorological wind. From this it may be assumed that a parallel air velocity a few tens of mm's above the cover is equivalent to a meteorological wind speed about 3 times greater. To check this, measurements were made under the solar simulator using the D.C. Hall collector, but with a non-selective black surface (Nextel) under stagnation conditions i.e. no fluid flow. Under these conditions $U_L/\tau a_e$ can be measured - see equation 5.27. $\tau a_C$ can be estimated using separate measurements of $\tau$ and $a$ so $U_L$ can be obtained. Several measurements of $U_L$ can be made for different wind velocities since $\tau a_e$ is independent of wind speed. The results are presented in Figure 5.47. No measurements could be made at zero wind speed because stratification within the room made it difficult to decide where to measure ambient temperature. Also plotted on Figure 5.47 is the predicted top loss for the collector using the steady state computer model (Section 4.2.1) which uses a meteorological wind speed to define the top heat loss. The same environmental
conditions were otherwise used in the model as in the experiment. From this it appears that the meteorological wind speed may be as much as 5 times larger than the measured wind speed in the laboratory. The discrepancy between these results and the results of Green et al (meteorological wind speed 3 times larger than in laboratory) [40] may be attributed to the type of wind generator used, ours producing a very turbulent layer next to the collector surface, to our measurement of the wind speed so close to the collector surface, and to a higher cover temperature in our case, because more radiation was absorbed in the cover than in the measurements by Green (performed with CSI lights).

This highlights the importance of more standardization of wind generators and the measurement of wind speed in solar simulators, in particular for collectors whose heat loss is dependent on wind speed such as non-selective collectors and in particular top-duct air-heating collectors where an increase in wind speed from 0 to 2.8 ms\(^{-1}\) can cause a 10% drop in efficiency. The direction of the wind may also be more critical for air collectors since the absorber temperature distribution is not uniform. Again this effect will be more important with plastic top collectors such as the SP collector.

Both the D.C.Hall and SP collector were tested indoors under the solar simulator. However, the D.C. Hall collector tested was with a non-selective absorber to avoid problems with the
spectral output of the solar simulator. The results of these tests are shown in Figures 5.48 and 5.49. The D.C.Hall collector results are compared with results predicted by a steady state computer simulation and the results for SP collectors are compared with outdoor collector testing. However, the outdoor comparison was not possible with this D.C. Hall collector, because the time was not available to test the D.C.Hall collector with the non-selective absorber.

From Figure 5.48 it can be seen that the indoor measurements on the structured polycarbonate collector correspond most closely to the measured efficiency outside under diffuse low intensity conditions. This is because of the low absorber temperature. However, the low intensity outdoor tests show a lower FRUL, and this is due to the large difference in wind speed and also the higher ambient temperature indoors which can have a substantial effect at low intensities [42]. The difference is not as large as may be expected, because these effects are partly offset by the higher sky temperature indoors (probably ± 40°C difference).

Figure 5.49 shows the results of indoor testing of the D.C.Hall collector. No outdoor results are available, because the outdoor tests were carried out with a selective surface. Thus for comparison, results from running RRDCT under the same environmental conditions as in the laboratory except with 5 times the measured wind speed (near the cover) and with sky temperature at ambient are presented. The
The computer predicted value of $F_{RUL}$ is smaller. When the collector was tested for leaks using a British Gas Meter, by blocking the fluid inlet with the meter in series before the fan, a leak of 1.36 Kg hr$^{-1}$ was detected corresponding to a 2% leak. From Figure 5.18 this leak corresponds to a 8% error in the measured value of $F_{RUL}$. So the corrected value of $F_{RUL}$ becomes 4.82. However, there still remains a slight discrepancy between the measured and predicted values, because one would expect the predicted value of $F_{RUL}$ to be higher due to the use of sky temperature at ambient. However, the wind multiplier is not very accurate, and experimental errors are slightly larger ($\pm 10\%$ error in $\eta$ for a $3^\circ$C temperature rise) for this type of indoor testing at low insolation levels because the percentage errors in temperature difference measurements are larger than at higher levels.

5.4.1. Conclusions about indoor testing

Indoor testing using a solar simulator presents a lot of problems in reproducing comparable results to outdoor testing. Even very expensive systems such as that at Cardiff need careful analysis of the results, and the cost of using such machines makes regular development work on air collectors impossible. There is therefore a demand for a somewhat cruder indoor test facility. Air collectors present their own particular problems in testing. Top duct collectors are very susceptible to heat loss from the wind
and therefore the particular wind generator and wind direction may affect how the laboratory wind speed relates to meteorological conditions. Great care must be taken in preventing air leaks in collector test facilities, because these lead to large errors.

Were the simulator to be reconstructed, it would be modified to improve its performance, as shown in Figure 5.50. It must be noted that nearly all the modifications would increase the cost of the test facility.

(i) The number of light sources would be increased by at least a factor of two, combined with the effect of bringing the lamps closer. This would substantially increase the intensity. Thus the collector performance will be more representative of real conditions and so improve the accuracy. This would mean a special air cooling circuit would have to be introduced. At £6 for each light source this would be a substantial increase in the cost of the test facility. However, if one compares this to the cost of a CSI lamp of £60 without its stator, the relative cost is small.

(ii) A larger more thermally stable room, or a ±0.1°C thermostatically controlled air conditioner, would be used. This would improve the stability of ambient temperature.
(iii) Two air cooled sheets of material would be introduced between the simulator and the collector as suggested by Kraus et al [43] to reduce the infrared radiation and the effective sky temperature. However, this would slightly reduce the intensity.

(iv) Axial fans would be used to provide a more natural wind profile.

(v) The collector would be run in a closed loop system. This would necessitate the use of an air reconditioner so that the collector could be tested at low air inlet temperatures. The advantages of this would be more stable inlet conditions and less heating of the laboratory.
5.5 Zero testing

The difficulty with outdoor ASHRAE steady state testing is the time for which the radiation must remain constant. The major problem with indoor testing is the cost of accurately obtaining indoors a solar spectrum. These problems can be overcome by only testing the collectors optical properties outdoors and its heat loss properties indoors. This type of testing has become known as 'zero radiation heat-loss testing' (or zero testing). The two factors describing the collectors operation are obtained from the Hottel-Whillier equation (equation 5.2) under limiting conditions.

(i) **Zero heat loss.** If the fluid inlet temperature is kept at ambient

\[
\frac{Q_u}{A_{c1m}} = F_R(\tau \alpha). \tag{5.53}
\]

This test is carried out outdoors under steady state conditions. This method does not strictly involve zero heat loss because the plate temperature is above ambient but this is taken into account by the factor \( F_R \).

(ii) **Zero incident radiation** With a collector indoors or at night time (if covered from the sky) there is zero net radiation exchange.
Readings of $F_{RUL}$ can be taken at several inlet temperatures.

These methods or similar methods have been proposed by several authors [44],[45],[46], and in May 1978 the German Bundesverbond Solarenrgie (BSE) working group completed and adopted 'Guidelines and Direction for Determining the Usability of Solar Collectors [47]. This uses combined indoor and outdoor testing by testing under zero heat loss outdoors and zero incident radiation indoors.

Good agreement between zero testing according to the BSE procedure and ASHRAE steady state testing for liquid collectors has been reported by Jenkins and Hill [48]. Similarly good agreement between theory and zero radiation testing for 14 different air heating collectors has been reported by Sheven and Hollands [49]. Smith and Weiss [45] have suggested that zero radiation testing is particularly suited for air collectors because the inlet temperature for the outdoor 'zero heat loss' test needs only to be kept at ambient. However, Wilkes [8] has reported that this method should only be used with extreme caution on air collectors, because during zero radiation testing at $T_i > T_a$ the collector is heated by the fluid, and for a given absorber plate temperature the rear duct is hotter than the absorber so you get a temperature inversion, whereas with the
radiative heating the converse is true. So for a given plate
temperature the heat loss through the back is greater under
zero radiation testing than under steady state testing such
as ASHRAE for DD77.

The importance of these effects has been examined by running
the computer program RRDCT (see Section 4.2.2) for the
collector described in table 5.3 with a 0.2 mm absorber plate
and duct back. The results of running RRDCT under ASHRAE
steady state conditions are compared with running the model
under zero radiation conditions and so obtaining a zero
testing efficiency curve. The same environmental conditions,
sky and ambient temperature and wind speed were used for both
runs. The results are presented in Table 5.8 and 5.9. From
these data the three efficiency curves most commonly used for
characterizing collectors [9] have been plotted. Figures
5.51, 5.52 and 5.53 show the efficiency plotted with respect
to the fluid inlet temperature, absorber plate temperature
and mean fluid temperature respectively. To generate the
efficiency curve with respect to the fluid inlet temperature
(Figure 5.51) you need to know $F_R(\tau_a)$ and $F_{UL}$. $F_{UL}$ is
determined from the zero radiation test for different fluid
inlet temperatures. The value of $F_R(\tau_a)$ is the same as that
generated from the steady state test with zero heat loss.
Likewise for the other efficiency curves (Figures 5.52 and
5.53) the same values of $(\tau_a)$ and $F_{ave}(\tau_a)$ are used as
obtained from the steady state test. These three efficiency
curves, although computer generated, represent the difference
that would be expected in real collector testing between steady state ASHRAE testing and zero testing, given that the same environmental conditions could be obtained indoors as outdoors.

The biggest difference between the steady state ASHRAE testing and zero testing curve is in Figure 5.51, when the efficiency is plotted with respect to the fluid inlet temperature. The difference is because for a given inlet temperature the value of $F_{RUL}$ is lower under zero radiation conditions because the average plate temperature is lower see Figure 5.54. However, at low inlet temperature the absorber temperature approaches ambient and the dominant heat loss is from the cover to sky temperature which is set at 20°C lower than ambient. As $F_{RUL}$ is quoted with respect to ambient and not sky temperature this gives an effectively larger value of $F_{RUL}$. The importance of sky temperature was investigated by running the computer model to simulate a zero radiation test for two inlet temperatures with the sky temperature 20°C less than ambient and the sky temperature at ambient - the simplest condition to achieve indoors an appreciable difference between the steady state and zero radiation tests can be seen (Figure 5.51), particularly at higher inlet temperatures. This problem can be overcome by using an environmental simulator [48] which simulates a low apparent sky temperature. However the sky temperature becomes less important at higher wind speeds. Jenkins and Hill [48] have shown no reduction in difference
between steady state ASHRAE testing and zero testing when altering the apparent sky temperature from ambient to $T_a - 16^\circ C$ while zero radiation testing indoors for liquid collectors tested with wind speeds $> 4 \text{ ms}^{-1}$.

The other two efficiency curves Figures 5.52 and 5.53 show better agreement because for a given absorber and mean fluid temperature the temperature profiles are similar for steady state and zero radiation — see Figure 5.55.

The BSE recommended the use of the average fluid temperature for their efficiency curves. But for a rear duct air heater the absorber and rear duct is always warmer with radiant heating i.e. ASHRAE testing than for zero radiation testing (Figure 5.55) and so the value of $F_{ave\, U_L}$ is different (Figure 5.57). Care should be taken in using the BSE guidelines for measuring $F_{ave\,(\tau\alpha)}$ for air heaters because they recommend that measurements can be made under steady state conditions with the average fluid temperature within $\pm 10^\circ C$ of ambient. If $F_{ave\,(\tau\alpha)}$ is measured with $T_m = 10^\circ C + T_a$ then $\frac{T_m - T_a}{T_a}$ will not equal zero but $0.014^\circ \text{Cm}^2 \text{ W}^{-1}$.

From figure 5.53 this can lead to a 4% error in $F_{ave\,(\tau\alpha)}$.

To obtain $F_{ave\,(\tau\alpha)}$ accurately $T_m$ must equal $T_a$. However in practice this is not easy to achieve because it requires $T_i < T_a$ and so some type of refrigeration must be used. Instead, it is proposed that $FR(\tau\alpha)$ be measured using an ambient fluid inlet temperature and that this value is converted using the following equation from Duffie and Beckman [9]
\[ F_{\text{ave}}(\tau a) = F_R(\tau a) \left[ \frac{MC_p}{A_c} + \frac{F_{\text{ave}} U_L}{2} \right] \]  

(5.55)

The value of \( F_{\text{ave}} U_L \) is obtained from zero radiation measurements.

5.5.1 Conclusions

All results reported of zero testing, when compared to the usual steady state testing (such as ASHRAE) for both liquid and air collectors, agree within the measurement errors associated with the ASHRAE standard of \( \pm 4-5\% \) efficiency points. Zero testing for rear duct air heating collectors is alright provided that temperature inversion is limited. This can be achieved by good heat transfer between the fluid and the plate, good radiative transfer between the plates of the rear duct and average heat loss from the front and rear of the collector. For zero testing top duct air heaters there are fewer problems as there is no temperature inversion.

Indoor measurements are easiest with a closed loop system, because the environmental conditions then remain constant and no indoor air conditioning is required. It is recommended that indoor tests only be carried out \textit{either} with wind speeds \( > 4 \text{ ms}^{-1} \) \textit{or} at lower speeds, but with the same sky temperature as normally experienced under steady state outdoor tests. If outdoor tests are to be performed at night time some measure of the sky temperature is recommended.
either by direct measurement [50] or by calculation [51]. Measurements for $F_{RUL}$ should only be made with $T_e - T_i > 5^\circ C$ to reduce errors.

Zero heat loss testing still has the disadvantage of requiring at least three hours of uninterrupted insolation around noon in order to obtain a value of $F_R(\tau\alpha)$. However, if the zero heat loss measurements is made under transient conditions this problem is overcome. Transient testing provides an efficiency curve for normal UK climatic conditions of diffuse low temperature operation. From the intercept of the efficiency curve with the efficiency axis an accurate value of $F_R(\tau\alpha)$ can be obtained. This value of $F_R(\tau\alpha)$ is independent of the climatic conditions under which the test is made (provided the diffuse and direct components of the radiation are accounted for), unlike the value of $F_{RUL}$. However, if the value of $F_{RUL}$ is obtained indoors under zero radiation conditions, then the collector can be tested both in the UK and the USA to produce the same efficiency curve. So a combination of transient outdoor tests and zero radiation indoor testing offers advantages of reproducibility compared to either of the two test methods individually.
5.6 Comparison of theory and test results

Figures 5.58 and 5.59 show the efficiency curves for the SP, and D.C. Hall (with selective absorber) collectors, tested by various methods. The data are summarised in Table 5.10. The figures also show theoretical curves generated from the steady state computer model described in Section 4.2. The figures clearly show the variation of results in different test methods and the difference between theory and practice is very large. To help identify the source of these differences the D.C. Hall collector was tested under stagnation. This gave $U_L/(\tau\alpha) = 5.9 \text{ Wm}^{-2}\text{°C}^{-1}$ for an absorber temperature of 130°C. This can be corrected to the absorber temperature during steady state testing of 80°C from figure 5.60 which shows a reduction of 0.4 Wm$^{-2}$°C$^{-1}$ for a reduction in absorber temperature from 130°C to 80°C. So $U_L/(\tau\alpha)$ is 5.5 Wm$^{-2}$°C$^{-1}$ at 80°C. But from the measured ASHRAE steady state curve $FRUL/FR(\tau\alpha) = 8.2 \text{ Wm}^{-2}\text{°C}^{-1}$. The difference can be explained by an air leak. Since there is no fluid flow under stagnation the effect of the leak should only effect the steady state flow measurement. From the leakage analysis this leak will not affect the value of $FR(\tau\alpha)$ but only $FRUL$. The % error in $FRUL$ is

$$\frac{8.2 - 5.5}{5.5} \times 100 = 50\%$$

From figure 5.18 this should correspond to a 9% leak = 6 kg hr$^{-1}$. A leak of 6-8 kg hr$^{-1}$ was measured using a gas meter. The leak was subsequently found around the edge of the
collector and was due to the weathering of the wooden support box on the underneath where re-sealing was impracticable. This shows the importance of leak testing and also how such leaks can be detected without a flow meter. If a leak of 6.8 kg hr\(^{-1}\) is taken into account the value of \(F_{RL}\) is still 1.0 Wm\(^{-2}\)C\(^{-1}\) too high. The emissivity of the selective surface was measured as 0.11 and of the cover 0.85 using a D and S Emissometer, these values were higher than those assumed in the computer model of 0.1 and 0.8. If the corrected values for emissivity are used and the radiation term doubled to compensate for the low sky temperature, the agreement between the theory and measurement is good.
5.7 Conclusions

The difference in the results for the different test methods shown in Figures 5.58 and 5.59 highlights the problem of which method should be adopted. The differences are due mainly to the different environmental conditions which the collector experiences, namely, diffuse and direct radiation, wind speed, sky temperature and ambient temperature. One could argue that the curve which is most representative of UK meteorological conditions during normal collector operation should be used.

Taylor [52] has examined the steady state efficiency of a collector installed at Cardiff (UK) operating as part of a solar hot water system. When this is compared with the steady state efficiency measured indoors according to the British Standard [30], see Figure 5.61, we see how unrepresentative the collector parameters measured under normal test procedures can be. This is because collectors in the UK spend most of their time operating at low efficiencies, with small temperature differences between the inlet and outlet and with irradiance values lower than the 800 Wm\(^{-2}\) recommended for British laboratory tests [30]. It is therefore proposed that tests be carried out at lower intensities which for outdoor conditions are normally transient and the most representative of UK conditions. A transient test curve provides values of \(F_{\text{RL}}\) and \(F_{\text{R}(\alpha)}\). \(F(\tau\alpha)\) does not vary much with environmental conditions,
except the proportion of diffuse to direct radiation which can be easily allowed for. However the value of $F_{RUL}$ is dependent on environmental conditions and it is this difference which makes it difficult to compare results of steady state testing with transient testing. So to get a 'standard' value of $F_{RUL}$ it is suggested that indoor zero radiation measurements are made. These are easily reproducible as the environmental conditions required are easy to control and so the test equipment required is cheap. Also the variation of $F_{RUL}$ for different environmental conditions can easily be characterised by measuring $F_{RUL}$ for different absorber plate temperatures and different wind speeds. So from the combination of transient and zero radiation testing an ASHRAE type steady state curve can be produced thus allowing comparison with American results, whilst also obtaining on route, a collector efficiency curve appropriate to U.K. conditions. This can all be achieved cheaply and without requiring steady state conditions.

Air collectors present particular problems in testing. They are more difficult to test using the ASHRAE method as they have longer time constants and so require longer test periods during which it is harder to ensure steady insolation levels. Their efficiency is more dependent on mass flow rate and wind speed, so they must be tested under more controlled conditions. Their efficiency is very dependent on air leaks and so some method is required to detect leaks. Transient testing must be carried out with data collected every $\frac{2\tau}{5}$.
Zero tests must be performed with care because the temperature distribution within the collector is different with zero radiation and pre-heated air flowing in, to when the collector is heated from a radiation source above the cover.

To summarize, it is suggested that steady state measurements are not made in the U.K., but that collector testing is done under transient conditions outdoors, and under zero radiation conditions indoors, and that several values of FRUL are quoted for different absorber plate temperatures. This is instead of testing according to the British Standard using a simulator with steady state conditions, or just outdoors under transient conditions at high levels of insolation.
6.1 What is a high performance collector?

The average insolation on a surface facing south at 45° to the horizontal situated at Kew (UK) is 3.53 GJm⁻² per year. Therefore, a 100% efficient collector at Kew (facing south at 45° to the horizontal) will collect the equivalent of £44 (1981) worth of electrical energy (at 4.48 p. a unit) in a year. This gives a rough guide to the maximum cost of an economically viable collector.

The steady state efficiency of a collector depends on fluid inlet temperature (see Chapter 5).

\[ \eta = F_R(\tau \alpha) - F_R U_L \left( \frac{T_i - T_a}{I} \right) \]  \hspace{1cm} (6.1)

When the fluid inlet temperature is at ambient the efficiency is dependent on the collector heat removal factor \( F_R \) and the product of the absorbance of the collector plate and the transmittance of the cover \( \tau \alpha \). The value of these three coefficients for collectors of competent design changes little with collector design. So, for a system which only requires a fluid inlet of close to ambient, such as swimming pools, there is little advantage in using a collector other than the simplest and so cheapest. However, at higher fluid inlet temperatures the operation of the collector is also dependent on the collector overall loss coefficient, \( U_L \).
This can vary over a wide range, producing the different performance characteristics of collectors - see Figure 6.1. A high performance collector is defined in this thesis as any collector whose efficiency is greater than that of a 'conventional flat plate solar collector' operating with high fluid inlet temperatures (> 40°C). So high performance refers to the operation of the collector at high temperatures. Figure 6.2 shows a conventional flat plate solar collector which consists of a matt black absorber and single pane glass cover.

High performance collectors are important because of the demand for high temperature heat in the industrial sector. For a space and water solar heating system with short term storage operating in the UK, a high performance collector can provide almost double the heat from a low performance collector [1] and for domestic heating utilizing inter-seasonal storage high performance collectors can substantially increase the energy collected - see Figure 2.23. High performance collectors are important for these uses, not only because they can collect more energy than a conventional collector at higher temperatures, but also because they allow energy to be collected at periods when conventional collectors could not operate, in particular during the winter period. Figure 6.4 shows the percentage of energy which falls above a threshold irradiance for each month. Thus for February 70% of the available radiation is above an irradiance of 165 Wm⁻². For every collector there is a
threshold irradiance below which no nett energy can be extracted. If we neglect the energy to transfer fluid within the system, the threshold level occurs when the incident energy absorbed is equal to the collector energy losses. In this case $\eta$ is zero. If $\eta$ in equation 6.1 is made equal to zero we get

$$I_{th} = \frac{U_L(T_i - T_a)}{\tau \alpha} \quad (6.2)$$

For example, a conventional collector with a selective absorber ($U_L = 4.3 \text{ Wm}^{-2}\text{K}^{-1}$, $\tau \alpha = 0.82$), operating with a fluid inlet temperature of $60^\circ\text{C}$, at an ambient temperature of $10^\circ\text{C}$ requires a threshold of 262 Wm$^{-2}$. That is, no nett energy is collected unless the irradiance is above 262 Wm$^{-2}$, and more if incident angle effects lower $\tau \alpha$. If we examine the same collector but now with a convection surpression device (a high performance collector) the threshold irradiance is 168 Wm$^{-2}$. From Figure 6.4 we can tell that the conventional collector with a selective absorber will collect no energy from November to February and only 25% of the available energy in October. Compare this with the convection suppression collector which can collect 72% of the available energy in February and 30% of that in November. When both the effect of increased steady state efficiency and increased operational time are taken into account, the high performance collector can collect 1.3 times the amount of energy that the conventional collector can collect. This compares with the increase in efficiency from the steady state equation of 1.1 times. So a greater increase in
efficiency is obtained from high performance collectors than is indicated by the steady state equation. What is more, the extra energy collected by the high performance collector occurs in the winter season when the demand for energy is higher. This in turn can affect the volume of storage required, particularly for interseasonal heat stores.

To summarise, high performance collectors are able to reach useful temperature levels in the winter when normal collectors can not. Figure 6.4 allows you to calculate how important this effect is in the UK.

Equation 6.2 shows that the threshold radiation is dependent on the value $U_L/\tau \alpha$. It has been suggested by Kenna [2] that the normalised loss coefficient $(U/\eta_* = U_L/\tau \alpha)$ is a suitable parameter by which to rate collectors, because it is a measure of the quality of energy that a collector can provide. However $\eta_*$ is needed independently to determine the collector efficiency. There seems little benefit then in replacing the two parameters $F_R U_L$ and $F_R \tau \alpha$ by two different parameters. Kenna admits that the normalised loss coefficient does not allow absolute collector comparison. Furthermore, there is the following danger in the normalised loss coefficient being adopted as a measure of the quality of a collector. The values of $U$ and $\eta_*$ have been adopted in the British Standard for solar heating systems for domestic hot water [3] for comparing the benefits of one collector to another. This may be acceptable for classifying flat plate
collectors, employing a liquid heat transfer media, for heating hot water in a single dwelling. But if solar collectors are to play a wider role in the UK than providing domestic washing water between spring and autumn, more information is required.

This chapter examines methods of improving collector performance and demonstrates that the two steady state collector parameters as measured by the British Standard [4] are not adequate to assess the merit of one collector compared to another when in operation in the UK at high temperatures. As yet, no algorithm exist like that for comparing domestic hot water systems (f-chart) [5] which means that very detailed modelling of specific systems is required.
6.2 Methods of improving the performance

The efficiency of a collector (equation 6.1) is dependent on its optical properties \((\tau \alpha)\), its heat loss \((U_L)\) and its ability to transfer heat to the fluid \((P_R)\). Also, the performance of a collector can be improved by changing the collector geometry, as in concentrating collectors. The efficiency of a collector in operation is also affected by the relative heat capacity of a collector, as this affects how the collector behaves in response to transients. This section reviews how these parameters affect the efficiency of operation of a collector, with particular reference to the UK. It is important to note that improving any one of these parameters may impair another.

6.2.1 Optical Properties

These are dependent on the properties of the absorber and cover with respect to electromagnetic radiation. In the ideal world we would have a cover transmissivity \((\tau)\) and absorber absorbivity \((\alpha)\) of unity at all angles, over all the solar wavelengths.

Table 6.1 lists the thermal and radiative properties for various real materials. In the UK float glass is normally used as a cover material as low iron glass is not commercially available. More recently thin plastics have become available which have very high transmissivities.
(Teflon, \( \tau = 0.96 \)). They are tough but not rigid, and so only tend to be of use for internal glazing. Thicker plastics such as polycarbonate and acrylic have become more popular for external covers because of their ease of handling and versatility. Their main disadvantages are that some have high transmission in the infrared, and that it still needs to be proved that they can sustain their advantages over long periods when exposed to the elements, in particular to ultraviolet radiation. However, rapid laboratory weathering now suggests that for some plastics this should be of little problem [6].

The absorbtivity of various collector materials is shown in Table 6.2. The material most commonly used for its high absorbtance, low cost, and stability at high temperatures, is Nextel Black Paint (3M) [7]. However, Nextel is not selective—see Section 6.2.2.

A great deal of research has gone into increasing the transmissivity and absorbtivity of materials. However, this has little effect on the performance of a collector at high temperatures. Figure 6.5 shows the efficiency curve of a standard flat plate collector with Nextel painted absorber and low iron glass cover and also the efficiency curve of the same collector except with \( \tau = \alpha = 1 \), the maximum theoretical improvement in \( \tau \) and \( \alpha \).
6.2.2. Overall heat transfer coefficient \( U_L \).

From section 4.2.1. the heat loss is made up of losses through the top, bottom and edge. The largest loss is through the top. The top loss is made up of four heat transfer coefficients \( h_{p-c} \), \( h_{r-p} \), \( h_{r-c-a} \) and \( h_w \).

The cover to ambient transfer coefficients \( h_w \) and \( h_{r-c-a} \) are dependent mostly on the siting of the collector. The values can also change appreciably with time, particularly in the UK where the combination of a windy and cloudy environment are the norm. Little can be done to affect the environment except for selecting sites which are sheltered from the wind.

The radiation loss from the absorber to the cover can be substantially changed, from that of a black absorber where \( \alpha = \varepsilon \) for all wavelengths, to a selective absorber where the emissivity at thermal wavelengths (infrared) is low compared to the absorptivity in the solar wavelengths. An idealised solar selective coating would have \( \alpha \) (solar wavelengths) = 1, \( \varepsilon \) (thermal wavelengths) = 0 see Figure 6.6. Table 6.2 lists the properties of various selective surfaces. Note that it is far more beneficial to increase the \( \alpha \) value by a few points than to reduce the \( \varepsilon \) value by the same number of points [8]. This is highlighted in a sensitivity analysis carried out by Makinen and Lund [9], where a change of 1% in the plate absorbtance (.95) resulted in a 0.25% increase in
solar fraction (% of annual energy supplied by solar) where as a 1% decrease in emittance (.95) results in only a 0.05% increase in solar fraction.

Selective surfaces are now easily available in the UK and at a low enough cost to make their application worthwhile even for low temperature operation. The most popular is Maxorb [10], as its durability and performance is proven, and its ease of application means that one producer can supply several manufacturers without the expense of transporting absorber plates to be coated. This allows large production runs without expensive transportation and so produces a cheap product. Costs for large quantities are £6 m⁻²(1980) [10]. Substantial research has gone into the production of selective absorber [11], and even selective paints [12] are available. Further large improvements in selective absorbers are unlikely. Figure 6.7 shows the steady state efficiency curves measured according to the British Standard indoors [4] of a flat plate collector with and without a selective absorber. Results from steady state tests of the same collector in operation outdoors in the UK [13] suggest that there is a discrepancy between the measured efficiency indoors under standard conditions and those obtained outdoors when collecting useful energy that is with lower intensities. This discrepancy is larger for selective absorbers than non selective absorbers.

If a completely selective absorber were possible, then the
only condition that the cover would need to meet is a value of unity for the transmittance at solar wavelengths. But this is not the case, so we require something on the cover to reflect the infrared radiation from the absorber back to the absorber - a selectively reflecting coating which transmits solar and reflects infra-red. A lot of work is at present being done to produce such infrared reflective coatings cheaply [14]. The best coating produced so far is an indium oxide layer on glass with a magnesium fluoride anti-reflecting film, giving a thermal emissivity $\varepsilon_t = 0.081$ and a solar transmissivity $\tau_s = 0.9$ [15]. But indium is a rare element, so costs would be high. A cheaper coating is a film of cadmium/tin deposited in an oxygen atmosphere to produce $\text{Cd}_2\text{Sn}_3\text{O}_4$. On soda glass this gives $\tau_s = 0.85$ and $\varepsilon_t = 0.2$, and an estimated manufacturing cost of £0.32 m$^{-2}$ [16] (1980).

Infrared reflective coatings are not available commercially in the UK, nor have any flat plate collectors been tested with such coatings in the UK. Infrared reflecting coating have however, been developed in the UK for heat reduction through double glazed windows [17][18]. However, these coatings are not suitable for collectors, as they reduce solar transmission substantially and the benefits of reduction in heat loss do not offset the loss of captured energy.

Section 7.1 reviews the methods of reducing the heat loss by reducing conduction and convection between the absorber and cover plate. Figure 6.8 shows the effect these various
measures can have on the efficiency curve of a flat plate collector. None of these improvements are available in commercially manufactured flat plate collectors, or have been tested under operation in the UK.

The costs of these various methods differ enormously and for some of the techniques a cost for commercial production can not be estimated because no commercial techniques exist as yet. In terms of proven simple technology the honeycomb appears to offer the greatest advantages. The production costs and efficiency in operation have been studied by Marshall et al [19]. They concluded that 'Lexan' (polycarbonate) is found to be the best performer for honeycomb applications using the 'expansion' method for hexagonal cell construction as it has excellent optical properties for honeycomb collector applications. However problems may arise with degradation at high temperatures. An aspect ratio \( \geq 5 \) (see Section 7.1.8) should be used to provide the best overall performance. The efficiency of a non selective flat plate collector was increased from 5% to 45% for 1 kWm\(^{-2}\) and an absorber temperature of 110°C. For a Lexan honeycomb the cost was £11 m\(^{-2}\) (1976), and estimated to be half if thinner Lexan sheets were available. More recently honeycombs made of 50\(\mu\)m thick polyethylene-terephthalate (PETP) have been costed at £8 per m\(^2\) (1983) [21] (mass 1 kgm\(^{-3}\)). This increases the performance of a flat plate collector to close to nearly that of an evacuated tube collector. Honeycombs do have the disadvantage of reducing
the transmission of radiation which is not normal to the collector - see Figure 6.9. This may be a particular disadvantage to operation in the UK where a substantial portion of the radiation is diffuse and so incident at angles other than normal.

The ultimate in convection suppression devices has been designed in Sweden, 'Silica Aerogel foam glass'[20]. This is a transparent glass (SiO$_2$) which has a thermal conductivity less than air (of 0.021 Wm$^{-1}$°C$^{-1}$), can withstand temperatures up to 750°C and is very light 0.05 grammes per cm$^3$. The material is however very brittle and so requires a cover plate. Figures for its transmissivity are however not available. Commercial production is being planned for the future but costs as yet are unavailable.

The ultimate theoretical reduction in convection and conduction is achieved through evacuation. This technique has been used with relative success on tubular collectors where existing technology for manufacturing fluorescent tubes can be utilized, and there appears to be no problem in maintaining a high vacuum ($<5 \times 10^{-5}$ torr) for periods up to 20 years provided the system is adequately baked out prior to evacuation [22]. Maintaining such a vacuum for flat plate collectors is more problematic. First the collector must be designed to prevent implosion. Initial tests at the Open University using 4mm glass supported with PTFE balls above the absorber, with a support area of less than 1% proved
successful. However after periods of evacuation of up to a week the glass was seen to crack. Maintaining a vacuum for prolonged periods with flat plate collectors is also more of a problem as sealing a squarer object is more difficult than a round object. However a company in Switzerland is currently developing an evacuated flat plate collector and have experienced no problems in maintaining a vacuum of $10^{-5}$ torr by utilizing a Zr/Al getter. The collector consists of 6mm float glass held 7mm above the absorber by spacers every 5cm. Operating temperatures of 300°C have been achieved [23]. One evacuated flat plate collector has also been produced commercially by Solarvak[24]. This utilizes acrylic as the cover and case of the collector. A cost of £94 m$^{-2}$ (1980) has been quoted. The collector is however only guaranteed for 1 year, and the company was not long in business. Work in Germany has estimated that flat plate evacuated collectors can be produced at costs 1.5 times that of a conventional single cover collector with a selective absorber [25].

A rather ingenious evacuated flat plate collector has also been developed in France [26] which utilizes existing technology by mounting flat absorber plates inside television (cathode ray) tubes. The advantage is that production lines already exist, the vacuum is guaranteed for 20 years and the cost is small (£20 (1981) per m$^2$). The disadvantages of this collector are that it is heavy (100 kg m$^{-2}$ of absorber), bulky (340 mm deep) and has a low $\tau\alpha = 0.68$. 
The use of gases heavier than air between the absorber and cover to reduce convection has also been investigated see Chapter 7. TNO in Delft [27] is also seriously considering commercial production of an evacuated flat plate collector utilizing xenon as a low conductivity residual gas. Test results show that a heat loss of 2 Wm$^{-2}$K$^{-1}$ at temperatures up to 150$^\circ$ are achievable. By utilizing a better selective surface it is anticipated that the heat loss could be reduced even further, to 1 Wm$^{-2}$ K$^{-1}$ (Figure 6.10).

Note:
(i) There is little benefit in reducing the heat loss from the absorber to cover due to convection and conduction below that of radiation or vice versa.

(ii) All the heat transfer coefficients vary with temperature, in particular the coefficients of radiation heat loss. So, where these dominate the overall heat loss, as is the case with evacuated collectors, the value of $U_L$ becomes a strong function of temperature. Therefore quoting a single $U_L$ value for some types of collectors is not representative of the collector under normal operational conditions.

(iii) Reducing the heat loss from the absorber to the cover by the methods mentioned above reduces the importance of the heat loss from the cover to the environment. This means that evacuated collectors with selective absorbers are more
important in the UK (one of the windiest countries in the world) than in less windy countries.

6.2.3 Collector heat removal factor, \( F_R \)

The value of \( F_R \) depends on several different properties of the collector, see Equation 4.19. However, the most important are the mass flow rate and the specific heat capacity of the fluid. Three fluid types are generally used to transfer the heat from the collector. They are water, oil and air in order of descending heat capacity. This means that if air is used a higher mass flow rate is required than for water. In practice the value of \( F_R \) changes very little with fluid flow rate for oil and water systems. However for air heating collectors the fluid flow rate is of critical importance to the performance of a collector, and the efficiency of various air heating collectors at various mass flow rates has been studied by Bhargava A.K., et al [28]. Figure 6.11 shows the measured change in collector efficiency with mass flow rate for the S.P. collector. The disadvantage of increasing the flow rate is that the energy required to pump the fluid around the collector system (parasitic power) is increased as the pressure drop across the collector increases, (see Figure 6.12). An optimum mass flow rate can be found for a particular system.

Figure 6.13 shows the theoretical system efficiency versus mass flow rate for a rear duct collector system operating
with a fluid inlet temperature of 60°C. The system efficiency is taken as the energy collected/m² minus the fan motor power/m² divided by the incident energy per square meter. For each square meter of collector a 1m length of 4" diameter pipe is assumed for the distribution system. The fan motor power required is assumed to be five times the theoretical power to maintain the flow. The analysis has been performed using the equations specified in Chapter 4, and has been carried out for three duct separations (2, 1 and 0.5 cm) and two levels of incident insolation (600 and 200 Wm⁻²). The optimum flow rate for 600 Wm⁻² appears to be ≈ 70 kg hr⁻¹m⁻² (for different collector types and systems the optimum may change). If this value is compared with the value of 20 kg hr⁻¹ used for a 3 cm rear duct in the Wimpey house [29] this explains the relatively poor efficiencies that they obtained.

From Figure 6.13 it can also be seen that the optimum flow rate changes with insolation. A similar change will occur with fluid inlet temperature. It is therefore suggested that a variable speed motor should be utilized to optimise an air system if it is to operate under varying conditions as in the UK. No air heating system with a variable speed motor has at present been operated in the UK, although a four speed control system has been tried out in the USA [30] and substantial annual energy savings predicted, especially in marginal solar climates.
6.2.4 Concentration

The efficiency at higher temperatures can be increased by decreasing the area from which heat losses occur. This is done by inserting an optical device between the source of radiation and the energy absorbing surface, to concentrate the level of incident radiation on the relatively small absorber.

Many designs have been set forth for concentrating collectors. Concentrators can be reflectors or refractors, they can be cylindrical or surfaces of revolution and can be continuous or segmented. Receivers can be convex, flat or concave and can be covered or uncovered. Many modes of tracking are possible. Concentrators can have concentration ratios from low values of 1.5 or 2 to high values of the order of 10,000. The greater the concentration ratio the greater the temperature at which energy can be delivered. However, greater precision in optical quality and positioning of the optical system is required to give high ratios.

Because diffuse radiation cannot be concentrated from an engineering point of view, concentrating collectors present a problem additional to those of flat plate collectors: they must be oriented to 'track' the sun so that the maximum beam radiation will be directed onto the absorbing surface so as to make up for the lost beam radiation.
Essentially the heat loss processes of a concentrating collector are the same as for a flat plate collector. However as the energy is concentrated, the temperature of the absorbing surface is higher and so the heat loss per unit area of absorber is greater. This in turn leads to the use of technologically more sophisticated methods of heat loss reduction such as evacuation and highly selective absorbers which can be utilized economically because the absorbing surface is relatively small in comparison to the collecting area.

Figure 6.14 shows the efficiency of a concentrating collector compared to that of a flat plate collector for beam radiation. The performance of concentrating collectors is high. However they have several disadvantages when compared to flat plate collectors.

(i) The construction of concentrating collectors is relatively complicated and must be carried out in factories and not on site

(ii) the collector can not form an integral part of the roof

(iii) they require a greater degree of maintenance

(iv) they can only concentrate the direct component of the solar radiation
(v) their orientation is critical.

For operation in the UK point (iv) is crucial, as approximately half of the radiation is diffuse. Figure 6.15 shows the % of total annual radiation month by month that is global and diffuse. Surprisingly, during the heating season (October to March) only 39% of the radiation is diffuse, compared to the rest of the year (April - September) when 70% of the annual radiation is incident and 52% of it is diffuse i.e. a larger proportion than in the winter.

In order to fully utilize direct radiation, concentrating collectors are normally designed to track the sun. A great deal of research has been directed towards optimizing the concentrator geometry to that all rays incident within a wide aperture angle will reach the absorber.

In the past it has been argued that conventional flat plate collectors collect most of their useful energy from direct beam radiation and so even if concentrating collectors do not track they out-perform flat plate collectors [31]. This is of course very dependent on the system that the collector is connected to. However even for domestic hot water systems a large proportion of the energy is collected with low intensity diffuse radiation [13], and this may be even more significant with interseasonal storage systems. It can also be argued that tracking the collector increases the annual
energy available to concentrating collectors to more than the energy available to a stationary flat plate collector. However, there is no reason why flat plate collectors cannot similarly be tracked.

There has been some modelling of the operation of concentrating collectors in the UK [32], however no firm conclusions regarding their usefulness were reached. There is also little operational experience of such collectors in the UK.

Work carried out in Portugal [33] suggests that a tracking concentrating collector will deliver more energy per annum than a flat plate collector at collector temperatures of more than 50°C above ambient (see Figure 6.16). However, this work is based on a location where a much greater proportion of the radiation is direct. Similar conclusions have been reached for work in the United States [34].

In Sweden,[35] where the diffuse component is slightly less than that in the UK (43% diffuse), measurements were carried out with a flat plate and tracking concentrating collector. It was found that 50% of the energy collected by the flat plate collector was actually from diffuse radiation. It was also found that a large proportion of the diffuse radiation occurred at low intensities, see Figure 6.17 (similar results are expected for the UK).
One of the advantages of high performance collectors is that they can operate at low intensities, because their threshold intensity is low. (see Section 6.1). If a large proportion of the low intensity radiation is diffuse, concentrating collectors will loose a substantial portion of their advantage. It has been shown that in Sweden [35], that at an average plate temperature of 58°C, a double glazed flat plate collector which is stationary ($F'U_L = 3.7 \text{ Wm}^{-2}\text{°C}^{-1}$ and $\eta_0 = 0.68$), out performs a tracking linear parabolic collector ($F'U_L = 1\text{ Wm}^{-2}\text{°C}^{-1}$ and $\eta_0 = 0.68$) if the tracking collector does not accept diffuse radiation. It is more likely that this result will be more applicable for the UK than the results from the U.S.A. and Portugal. It also appears unfair to compare a tracking concentrating collector with a non tracking flat plate. In Sweden [35] it has been shown that a tracking flat plate collector can compete with a tracking concentrating collector at all inlet temperatures below 100°C, which rules out the operation of tracking concentrating collectors with water as the heat transfer fluid, unless a pressurised system is considered or oil is used as the heat transfer fluid with the cost of an additional heat exchanger.

Systems utilizing concentrating collectors also require a greater proportion of oversizing because the fraction of direct radiation we get each year varies more from year to year than the amount of total radiation. The coefficient of variation of recorded direct radiation is about 5% at Kew
compared to a value of 3% for total radiation [36].

The potential for concentrating collectors in the UK appears to be limited to temperatures greater than 100°C. However, a full assessment has not been carried out in the UK for concentrating collectors operating at different temperatures and with various heating demands.

6.2.5 Effective Thermal Capacity

So far we have been looking at methods of increasing the instantaneous efficiency of collectors. If the collector had zero mass or the conditions in which it operated were steady, then the instantaneous efficiency curve would represent the operation of the collector. However, all collectors do have mass and the climatic conditions in the UK do fluctuate a tremendous amount. A collector's response to these fluctuations depends on its capacitance and thermal resistance and (under flow) also on flow rate. A large proportion of the work done on collectors has concentrated on the instantaneous or steady state coefficients to increase the performance, with very little on the effect of transient operation. This is partly due to the majority of solar work being based on the experiences of the USA, where more stable climatic conditions prevail and the adoption of simple flat plate solar water heaters as the major application of solar energy within the UK. The reduction in thermal capacity for solar water heaters is more difficult to achieve as the absorber
must be able to support the heat transfer fluid and it is the absorber's mass which is the principle element in the operation of the collector. Therefore the majority of collectors in the UK have been of the same thermal capacity with little potential for reduction. However with the potential for using air heating systems in the UK, and also in examining the use of concentrating collectors, we meet types of collector which can be designed with substantially different thermal capacities. It is important to understand what, if any, improvements can be achieved in the performance of a collector by reducing its thermal capacity.

There are two distinct transient characteristics of solar radiation, there is the diurnal variation due to the axial spin of the earth and the shorter term fluctuations due to the weather.

In France [37] a large proportion of the high level radiation has been reported to occur over very short periods (< 10 minutes). Similar conditions are expected to occur in the UK. If the insolation is continuously varying on a thermally massive collector operating at a high fluid inlet temperature, the absorber surface may never warm up to the desired fluid temperature, and so no useful energy can be collected.

To study the effect that mass has on a collector, the transient model described in section 4.2.2. was run for air
collectors differing only in their thermal capacity. Results were reported by Jones and Oreszczyn (Appendix C) for an air heating collector operating over a day under different conditions of varying insolation.

The collector investigated is a flat-plate rear-duct air heating single cover collector designed to give good performance. The collector is oriented south, and the simulation is run for a simulated day corresponding to various times in the year.

The collector model is run over a day with a constant fluid inlet temperature as would be the case if it were operating with a large store (of the order of several days storage). The air flow rate in the rear duct is zero until a specified minimum power would be extracted from the collector were the air to flow. The air then flows at a constant rate. Flow ceases if the power extracted falls below this minimum value.

Five collectors were modelled with time constants with fluid flow varying from 85 to 1400s. This corresponded to a collector with absorber and duct-back thickness of between 0.2 mm and 5.0 mm (see Figure 5.35). Fluid inlet temperatures considered were 30°C and 50°C.

The model was run with simulated weather corresponding to 21 June, 21 March and 21 December (See Figure 6.18). Wind speed
of 1 ms\(^{-1}\) constant was assumed and the insolation varied through the day. It could be diffuse or direct, and in diffuse conditions an alternation between high and low levels with periods of 600s and 1200s could be superimposed on the diurnal variation.

For each 'day' an average thermal efficiency was obtained, defined by

\[ \eta = \frac{\text{total energy extracted by the air flow on the day}}{\text{total insolation incident on the collector over the day}}. \]

A day was the time from sunrise to sunset. The efficiency has been plotted in Figure 6.19 against \((T_i - T_a)/I\), where \(T_i\) is the constant fluid inlet temperature, \(T_a\) and \(I\) the arithmetic mean values of ambient temperature and insolation for the period sunrise to sunset, the results being coded in accord with Table 6.3.

At each value of \((T_i - T_a)/I\) in Figure 6.19 there is a column of results, and in every case the collector with the smallest thermal capacity is at the top and the largest thermal capacity at the bottom. Clearly the lower the thermal capacity the better the performance. The gain in efficiency from the heaviest to the lightest is slightly less than the gain obtained when a selective coating is inserted on to non selective flat back collector.
The general improvement with reducing thermal capacity arises because with a diurnal envelope the slower warm-up of a high mass collector in the morning is not compensated by the slower cool-down in the afternoon.

The intermittent insolation only has a marginal effect; a low mass collector will 'follow' the insolation, possibly switching the air flow on and off, a high mass collector, once it has warmed up to the point where the air flow switched on, will tend to stay at a fairly constant temperature. The overall effect, for a wide variety of conditions, is that the time averaged temperature of the air flow are not very sensitive to the mass, therefore there is very little difference in the amount of heat extracted. A similar conclusion was reached by Klein et al. [38].

Also plotted on Figure 6.19 is the steady state efficiency curve obtained by running the computer model under simulated conditions as specified in the ASHRAE standard [38]. Under steady state conditions a similar curve is obtained for all thermal capacities. Note that $\eta$ and $\bar{\eta}$ are different, even for the low thermal capacity collector (whose response is close to a zero capacity collector). This is due partly to the steady state test being performed with normal insolation, whereas in the transient diurnal efficiency the sun's angle changes. And it is partly due to the average insolation in marginal conditions being less effective than the maximum insolation in diurnally varying conditions that give the same
average.

Not only does the daily efficiency improve for collectors with a lower thermal capacity but also lower capacity collectors provide a higher fluid outlet temperature: of the order of 10°C when comparing the lightest with the heaviest collector. Depending on the demands of the load this may be an important advantage for low thermal capacity collectors. The difference in daily efficiency between the lightest and heaviest collector is greater for higher inlet temperatures. High performance collectors are more likely to operate at higher temperatures so the thermal capacity will be more important with high performance collectors.

Collectors have two time constants. The response to changes when the collector is stagnating (no fluid flow) and the response when fluid is flowing. The latter is the more commonly measured value as it forms part of the testing standards [39]. Note that this value changes with fluid flow rate. Generally the time constant for air collectors is less under stagnation than with air flow (see Figure 5.35). Also, the stagnation time constant for air collectors is less than for water collectors, the reverse being true under flow conditions.

Large stagnation capacities have been shown to be a problem with some water heating collectors in the UK [40] as they show a slow response to incoming radiation due to the
thermal capacity and conductivity of the fluid stagnating in the collector and the adjoining pipe work. This means that insolation incident for short periods is not successfully collected.

Some manufacturers of air heating solar collectors are using the low thermal capacity nature of their collectors as a selling point for transient climates where they can capture useful heat during brief periods of winter brightness [40]. To be able to predict the importance of this it is necessary that standard tests are carried out on collectors to measure their time constants under stagnation, and also that more information is made available about short term fluctuations in insolation. At present most simulations are carried out on hourly weather data as this is the only readily available data. However, if collectors have time constants much shorter than an hour this will not be adequate.
6.3 Conclusion

Methods of increasing the performance of collectors have been investigated. Reducing the heat loss from absorbers to cover is found to offer the greatest potential for improvement in performance. Concentrating the incident radiation is shown to produce similar improvements, but because \( \approx 50\% \) of the radiation in the UK is diffuse, their application is probably limited to the supply of heat at temperatures greater than 100°C. Since they cannot be maintained on the roof their application in the domestic sector is not envisaged.

Experience of operating concentrating collectors in the UK is limited as is detailed data on the relative proportions of diffuse to direct radiation. Several methods for reducing convection and conduction between absorber and cover for flat plate collectors have been identified. However experience of operating such systems in the UK is very limited. The most favourable appear to be the use of a honeycomb structure and evacuation and these are dealt with in greater detail in Chapter 7. Until data are available as to the proportion of non normal incidence radiation it is difficult to predict whether the limited transmissivity of honeycombs at angles other than normal to the collector will be a problem for operation in the UK. The reduction of \( h_r p-c \) is more important in windy climates such as the UK.

The standard parameters \( F_{RL}^U \) and \( F_{RTa} \) have been identified as not being the only parameters which affect the performance of
a collector, particularly in climates such as the UK where intermittent conditions are the norm. The thermal mass of the collector, both with fluid flow and under stagnation, have been identified as having an effect on the performance, lighter collectors collecting more energy than heavier ones. Data on the intermittent nature of insolation in the UK is very limited. It is suggested that measurement of both the thermal time constants of collectors under stagnation and with fluid flow be incorporated into collector testing standards.

So far most of the research in the UK has been directed at increasing the thermal performance of collectors operating under steady state conditions. Further improvements are limited. It is suggested that more effort should now be put into research into collector operation under UK conditions, that is, under transient and diffuse conditions, and in particular examining the control strategy for systems. Other countries' results on strategy are of little meaning to us whereas results for improving $\tau$, $\alpha$, $U_L$, $F_R$ etc will apply to any country, so if research funds are limited then they should be invested in the software, as we can import the hardware.

Testing of collector operation in the UK has been predominantly on the standard flat plate collector type. The preceding analysis has however identified collectors with low heat loss, high acceptance of diffuse radiation, low thermal
capacity and transmissivity over all angles of incidence. More experience of operating such collectors in the UK is needed to assess their advantages and disadvantages. One collector in particular has been identified as possessing the above properties and so is seen as being ideal for the UK. However, it has been developed in the USA by the General Electric Company [42] (see Figure 6.20). This is an air-cooled solar collector. It has a phenolic polymer structural foam body, a black painted fibreglass mat absorber and a coplanar, parallel, closely packed array of evacuated cylindrical glass tubes for the collector cover (fibre mat tube collector, FMTC). The estimated cost of this collector is £45m⁻² (1982). The advantages of the FMTC are that even at low pressure drop its porous matrix absorber allows a higher heat transfer ($F' = 0.99$ when $m = 150 \text{ ls}^{-1} \text{ m}^{-2}$) from absorber to fluid than with a plate absorber, a high level of absorptance 0.97 due to the fibre absorber acting as a cavity, and a very low top loss due to the evacuated tube cover, which eliminates convection. However, breakage has been a problem with tubular collectors [43] and therefore the use of a honeycomb structure between the cover and absorber may have more effective heat loss reduction. However, the collector would then lose out on its very high angular transmissivity which the tubular cover provides, (see Figure 6.21). This means that the FMTC only has a 10 per cent drop in energy absorbed for diffuse compared to direct and so is ideal for capturing solar energy in the cloudy UK. The FMTC also has a fast response (stagnation time = 16 minutes)
because of its low thermal mass, which also makes it attractive for the UK climate. The measured and theoretical steady state efficiency of FMTC is shown in Figure 6.22. The efficiency improves with a reduction in intensity of the radiation. This is quite important for the collector's operation in the UK because the standard test curve ($I = 800 \text{ Wm}^{-2}$) would correspond closely to the curve for $1025 \text{ Wm}^{-2}$, whereas the majority of the collector's operation in the UK would be at lower intensities and so the collector would operate more efficiently than the standard steady state tests would suggest. The large improvement in efficiency with a reduction in intensity is because the overall collector heat loss is predominantly radiant, which is very temperature dependent. Since lower levels of insolation produce lower absorber temperatures this results in a greater efficiency. The radiant heat loss is primarily from the inner to the outer edge of the evacuated tube; this could be reduced by inserting an infra-red reflecting film within the tube.

For comparison Figure 6.22 also shows the efficiency of a conventional flat plate collector with selective absorber and the associated improvement in its efficiency due to a reduction in intensity. If one compares the high insolation curves for both the collectors, that is the curves which would be obtained under standard steady state testing, one would not expect the FMTC to collect up to twelve times the energy during a day's operation under low intensity operation that modelling has shown [42]. This collector provides a
classic example of how the standard steady state, direct radiation, high insolation test, such as adopted by the British Standard does not do justice to the collector operation in the UK. This example should encourage the adoption of testing standards which go further towards predicting all day or all year performance in the UK. Although the present standards may be sufficient for the solar collectors presently in use in the UK, this is not the case for high performance collectors.

There is a need to replicate the test results for the FMTC by testing under transient conditions and indoors as suggested in Section 5.7. This is because the present results suggest a very high efficiency. If these results are replicated then a full scale comparison between the energy collected by a conventional flat plate collector and an FMTC should be undertaken in the U.K. for the period of at least one year and the difference in energy consumption analysed.

No detailed cost comparisons have been carried out in the previous analysis to assess the relative benefits of one technique when compared to another. Costs are very dependent on production capacity, and very few of the techniques described are commercially available and so estimation is very difficult. The benefits are also very system dependent and so costs are only worth calculating for a particular system. There is therefore a need to perform several full scale tests with low thermal capacity high performance
collectors attached to domestic heating systems and to compare the results against those obtained from normal steady state collector testing and conventional flat plate collectors.
Chapter 7 Convection and conduction reduction between absorber and cover to improve collector performance

The development of a high performance collector requires that the heat transfer due to convection and conduction between the absorber and the cover plate is at a minimum. This chapter reviews the theory of convection and conduction between two parallel plates and examines the potential for increasing collector efficiency by reducing the pressure, substituting gases, and inserting convection suppression devices (CSD) between the absorber and cover of a flat plate collector. A simple method and results are also presented for verifying theoretical predictions for collector top loss.
7.1 The theory of convection and conduction heat loss

7.1.1 Conductivity in gases (k)

Heat is conducted (i.e. thermal energy is diffused) in a gas by the random motion of molecules. Higher velocity molecules from higher temperature regions move about randomly and some reach regions of lower temperature.

By a similar random process, lower-velocity molecules reach higher-temperature-regions, thereby net energy is exchanged between the two regions. The thermal conductivity depends upon the space density of molecules, upon their mean free path, and upon the magnitude of the molecular velocities. Kinetic theory gives the following expression for the thermal conductivity of a gas of rigid non-attracting spherical molecules

\[ k = \frac{1}{3} \rho \lambda c_v \bar{c} \]  

(7.1)

The coefficient of viscosity is

\[ \mu = \frac{1}{3} \rho \lambda \bar{c} \]  

(7.2)

which on substitution gives

\[ k = c_v \mu = C_v \mu/M \]  

(7.3)

where \( C_v \) is the molecular heat at constant volume,
\( M \) is the molecular weight.

Since \( \rho \propto \rho, \lambda \propto \frac{1}{\rho} \) and \( c \propto \sqrt{T} \) where \( \rho \) is the pressure and \( T \) the temperature of the gas, it follows that \( k \) is independent of pressure and proportional to the square root of the absolute temperature. In practice equation 7.3 can be rewritten

\[
k = i c v \mu \quad (7.4)
\]

Where the value of \( i \) for monatomic gases is 2.5 and for diatomic gases is 1.9.

Figure 7.1 shows a plot of thermal conductivity of various gases at atmospheric pressure versus molecular weight. This shows that in general the heavier the molecule the smaller the thermal conductivity of the gas.

**Conductivity in Rarefied Gases**

According to equation 7.1, \( k \) does not change with pressure since as the pressure is decreased the density is decreased but the path of the molecules (\( \lambda \)) is increased in proportion

\[
\lambda = \left( \frac{R}{\sqrt{2} \pi d N_A} \right) \left( \frac{T}{P} \right), \quad \text{(For air at } 20^\circ\text{C } \lambda = \frac{5}{\rho} \text{ (cm)} \right)
\]

where \( p \) is in millitorr). \( (7.5) \)

Now consider the case of conduction between two surfaces at different temperatures, when the mean free path (\( \lambda \)) of the molecules is less than the dimensions of separation between the two plates. (This is known as the free molecule
conduction regime. Surface molecules then transport heat across the whole apparatus at a single bound so that the whole of the gas must be assumed to be at a uniform temperature; there is no longer a gradual temperature gradient, as there are no inter-molecular collisions. One would then expect a molecule to leave a surface with the energy appropriate to that surface temperature and transfer that energy to the other surface. But this proves to be incorrect, and the energy transferred is substantially less, as though the interchange of energy between the solid wall and the absorbed molecules of gas were far from complete.

This led Knudsen to introduce a quantity (a) which he described as a coefficient of accommodation. He imagines in brief that when a molecule is absorbed by the wall, its energy change does not correspond to the whole range of temperature difference \( \Delta T \), but only to a fraction (a) of this difference. This value (a) is found to depend on the molecular weight of the gas, the molecular weight of the surface material, the temperature and other physical conditions, such as, the cleanliness of the solid surface. Almost all values from 0 to 1 appear to be possible. A crude relationship for (a) is

\[
a = \frac{4 m m'}{(m + m')^2}
\]

(7.6)

where \( m \) = wall molecule mass and \( m' \) = gas molecule mass.

This theory does not apply to complicated molecules (see
Recently the accommodation coefficient in the free molecule conduction regime has been measured for an assortment of collector materials and the accommodation coefficient was seen to decrease with surface temperature [1]. This was attributed to smaller interaction times at higher temperatures. This may account for the fact that the measured thermal conductivity between two evacuated surfaces does not increase over a temperature range 100 to 200°C [2], as would be expected because of the increase in pressure due to the heating. Instead the increase in conductivity caused by the pressure rise has been offset by the decrease in accommodation coefficient. So the overall thermal conductivity is close to being independent of temperature.

The apparent conductivity between two plates at low pressures where $\lambda >> s$ is called the Pirani range and is characterised by

$$h_c = \frac{k}{s} = \frac{a}{(2-a)} \left[ \frac{(8-1) c_v p}{(8\pi r T)^{1/2}} \right]$$

(7.7)

### 7.1.2 Convection

When a heated body is in contact with a gas or liquid medium having a lower temperature - or the converse, heat is transferred between the surfaces and the fluid. The heat
change causes a change in density of the fluid. This change is large for a gas being expressed by the gas law, so the positive buoyancy of the gas adjacent to the heated surface causes it to move upwards. The mass motion of the fluid adjacent to the surface is termed convection. Because the buoyancy force is always upwards, convection only takes place when the hotter body is below the cooler.

**Natural or Steady State Convection**

When there is no external driving force like wind or pumped flow, natural convection can take place. This means that a balance between the buoyancy effects caused by a change in density with height and the viscosity of the flow medium occur. The force generated by the density difference is exactly balanced by the force generated by the viscosity, the velocity of convection being the variable generating the viscosity force. In the mathematical description of convection the usual treatment is to use dimensional analysis to arrive at the appropriate mathematical forms, allowing arbitrary constants to yield the correct answer in a limited domain of applicability. It is important therefore to understand the conditions under which these empirical equations hold.
The usual dimensionless groups used for convection studies are

Nusselt number (Nu) \[ = \frac{h L}{k} \] (7.8)
Reynolds number (Re) \[ = \frac{\rho v L}{\mu} \] (7.9)
Prandtl number (Pr) \[ = \frac{\mu c_p}{k} = \frac{\nu}{\alpha} \] (7.10)
Grashof number (Gr) \[ = \frac{g \beta \Delta T L^3 \rho^2}{\mu^2} \] (7.11)
Rayleigh number (Ra) \[ = \frac{\rho^2 g \beta \Delta T L^3}{\mu k} \] \[ c_p = \text{Gr.Pr} \] (7.12)

where \( L \) is the minimum dimension between the hot and cold walls.

**Enclosed fluid between two Flat Plates**

The heat transfer coefficient \( h \) is defined as follows

\[
\frac{Q}{A} = h \Delta T
\] (7.13)

For horizontal air spaces there are two different cases. If the upper plate is at the higher temperature, no convection effects will arise, except possibly at the edges, and heat transfer will be entirely by conduction. The heat transfer coefficient \( h \) is then merely \( k/s \) where \( s \) is the distance between the two flat plates. Therefore the Nusselt number is 1.0.

If the lower plate is warmer, an unstable condition arises, lighter layers of fluid are overlain by denser layers.
According to theory a Grashof number based upon s, less than 1700 results in no motion and the simple conduction rate pertains.

For greater values, natural-convection effects arise.

Jakob, M [3] has correlated the data of various investigators as follows

\[
\begin{align*}
\text{Nu} & = 0.195 \sqrt[4]{\text{Gr}} \quad 10^4 < \text{Gr} < 4 \times 10^5 \quad \text{(7.14)} \\
\text{Nu} & = 0.068 \sqrt[3]{\text{Gr}} \quad 4 \times 10^5 < \text{Gr} \quad \text{(7.15)}
\end{align*}
\]

The lower range (Equation 7.14) corresponds to an ordered, cellular convection process, as in Figure 7.2. This can be observed with the help of an infrared camera - see plate 7.1. Oil is placed inside a dish which is heated from underneath - Figure 7.3. To obtain uniform heating the dish is placed inside a secondary dish of oil. This dish forms the lower plate and the surface layer of oil the top plate. The infrared camera sees the surface temperature of the oil. Hotter spots are displayed as being lighter. If a structure of smaller dimensions than the convection cell is inserted then convection can not take place. This is the principle behind convection suppression devices such as honeycomb s - see Section 7.1.8. This method of observation can provide a useful tool in examining the usefulness of such devices.

More recently Hollands has updated the work of Jakob and extended it to include inclined layers [4]. But instead of
using Gr, Hollands had used Ra.

For horizontal plates, pure conduction exists provided Ra is less than a critical value of $Ra_c$ i.e. for $Ra < Ra_c$, $Nu = 1$.

The behaviour of the heat transfer in the immediate vicinity of the critical condition is known on theoretical grounds to be

$$Nu = 1 + k \left[ 1 - \frac{Ra_c}{Ra} \right] \text{ for } Ra > Ra_c \text{ and } 1708 < Ra < 10^4 \quad (7.16)$$

where $k$ is a constant.

For horizontal plates $Ra_c$ is 1708 and $k$ is 1.44 [5]. Hollands has tested equation 7.16 for pressures below atmospheric and observed the relationship to hold for air to within 3% of the average experimental value.

For inclined plates there exists fluid motion for any finite Ra. However provided Ra is sufficiently small, the fluid motion consists of one large cell (Figure 7.4), called the base flow, with the flux rising near the hot surface, falling near the cold surface, but with the streamlines parallel to the boundary surface everywhere except at the extreme ends of the enclosure where the fluid turns. The heat transfer in this regime is consequently upwardly conductive ($Nu = 1$) except at the extreme ends where there is some convective heat transfer associated with the fluid turning - see Figure
7.5. However, as the aspect ratio (= H/L, Figure 7.4) of the plates becomes increasingly large the contribution of this convective heat transfer to the average Nusselt number for the slot becomes vanishingly small so the average Nusselt number approaches unity. This conductive regime exists for air provided the Raleigh number is less than the critical value $Ra_C$ given by

$$Ra_C = \frac{1708}{\cos \theta} \quad (7.17)$$

Consequently we have

$$Nu = 1 \quad \text{for} \quad Ra < \frac{1708}{\cos \theta} \quad \text{where} \quad \theta < 70^\circ$$

Values of Nu for an inclined surface have been measured for $Ra > Ra_C$ by Hollands [5]. The experimental method consisted of maintaining a constant temperature difference and spacing across a tilted air layer and passing through the point of instability by slowly increasing the air pressure by steps and measuring the heat flux at each step. The heat flux was measured by a combination of a heat flux meter and guarded hot plate principle - see Section 7.2. The results provided the following relationship for air.
\[ \text{Nu} = 1 + 1.44 \left[ 1 - \frac{1708}{\text{Ra} \cos \theta} \right]^* \left[ 1 - \frac{(\sin 1.8\theta)^{1.6} 1708}{\text{Ra} \cos \theta} \right] + \left[ \left( \frac{\text{Ra} \cos \theta}{5830} \right)^{1/3} - 1 \right]^* \]  

(7.18)

where * brackets go to zero when negative, that is if \( \text{Ra} \cos \theta \) < 5830 the term in the second squared bracket becomes zero, and if \( \text{Ra} \cos \theta \) < 1708 both terms in square brackets becomes zero so \( \text{Nu} = 1.0 \).

Equation 7.18 is found to fit data using air from tilt angles 0 < \( \theta \) < 60° and 0 < Ra < 10^5 to an expected maximum error of about 5% for the Nusselt numbers, and it can be used up to 75° but with error of up to 10%. It is also expected to be valid for Ra > 10^5 but this has only been tested for \( \theta = 0 \).

7.1.3 Plate-cover separation(s)

As you increase the cover separation you eventually reach the point where \( \text{Ra} > \text{Ra}_c \) so you are no longer in the pure conduction range \( \text{Nu} = 1 \) (see Figure 7.6). The separation at which this occurs is dependent on the temperature. As the spacing increases you then enter the initial convection region II as represented by equation 7.18. This initial increase is followed by a decrease in the value of \( h \), thus leading to a peak at b. In region IV the mode of heat transfer is turbulent convection. This region is represented by a very gradual decrease of conductance with gap spacing. At a certain point c on the curve the heat transfer rate
falls below the previously attained minima at (a). Since the minimum (a) shifts with temperature it is important not to use this minimum when constructing a collector which operates over a wide range of temperatures, it is far better to construct it at a gap spacing bigger than c. Figure 7.7 shows values of $h$ for air, varying $s$ and $\theta$ at $T_{\text{cold}} = 275K$, $T_{\text{hot}} = 325K$ and $10^5$ Pa. The best gap spacing for a collector is therefore $> 10$ cm. This also applies for different gases, temperatures and angles of inclination. However, there is little improvement beyond 8 cm. Increasing the collector cover to absorb spacing however increases the collector material costs and the side shading of the collector. A spacing of 5 cm is therefore a good compromise for most collector applications.

Buchberg. H. et al [6] has calculated that for $s = 1.6$ cm (equivalent to point (a) Figure 7.6 at $\Delta T = 30^\circ C$) the efficiency of a single glazed collector with a selective absorber is 47.6% while a spacing of 5.08 cm (equivalent to c) gives an efficiency of 53.6%. This would mean that the area of collection could be reduced by 12.6% and emphasises the importance of designing collectors with modest gap spacings.

7.1.4 Angle of tilt

As the angle of tilt increases so does $R_a$ (Equation 7.17) so for tilted collectors the onset of convection occurs at
higher temperatures. Also when \( Ra > Ra_c \), \( Nu \) is smaller for tilted collectors than horizontal (see Equation 7.18), thus the heat transfer is less for tilted collectors, this holds for all gases - see Figure 7.8. An approximate rule is a tilt of 70° to horizontal gives \( \approx 75\% \) of the heat transfer the collector has in the horizontal.

7.1.5 Temperature

It is important for solar collector operation to note that \( h_c \) is a function of temperature - see Figure 7.9, and that not only is it dependent on the temperature difference but also on the absolute temperature. This has important consequences for making predictions regarding collector performance. When making calculations of the efficiency of a collector you must assume a value of \( h_c \), if the collector is to be operated over a small temperature range or \( T_i > 140^\circ \text{C} \) it is safe to assume a single value of \( h_c \). If it is to be operated over a large range of temperatures or very small temperature differences close to ambient, a single value of \( h_c \) is not justifiable. For example if a collector operates over a temperature range of 140°C, and the only value of \( h_c \) is assumed that corresponding to 140°C, the heat loss at 40°C will only be 70% of that predicted (see Figure 7.10). In particular this will have an effect in the expected warm up time of the collector. Figure 7.9 also shows the variation of \( h_r \) (see equation 4.7) with temperature, for a selective and non-selective absorber, with a glass cover. This shows that for
a collector with a non-selective absorber radiation is the dominant form of heat loss from the absorber to the cover, but that for a selective absorber convection and conduction dominate. Also for a selective absorber the absolute change in $h_T$ with temperature is small compared to the absolute change in $h_C$.

7.1.6 Gases other than air and mixtures of gases

Utilization of gases other than air in the collector can reduce the heat transfer coefficient loss so long as:

1. The gas thermal conductivity is less than that of air - see Figure 7.1.

and

2. The effective Rayleigh number is similar to that of air or less for a given geometry - see Figure 7.11.

From Figure 7.1 you can see that the thermal conductivity of various gases is dependent on molecular weight, but the effect on the Rayleigh number of molecular weight (figure 7.11) is not so obvious. This is due to the variation of viscosity with the molecular structure. The heat transfer coefficient for different gases and their molecular weight are shown in Figure 7.12. Note that for many gases the reduction in conduction for high molecular weight gases is almost completely offset by the increase in convectivity. Figure 7.12 has been plotted assuming that equation 7.18 holds for all gases whereas it has only been experimentally
verified for air, argon and carbon dioxide [7].

Xenon provides the largest reduction in heat transfer but unfortunately both xenon and krypton are far too expensive (at today's prices) to be used in flat plate solar collectors (Figure 7.13). Therefore unless these costs drop dramatically, their application in the solar field is limited, although they might have some applications in concentrating collectors where the absorber area is smaller and so the volume surrounding it will also be small. There is also the possibility of using xenon at reduced pressure. Xenon is very expensive at present because the demand is only for very small and very pure quantities. However, since it is a constituent of the atmosphere, it should be possible to produce the impure gas much cheaper. At present argon looks to be the most promising gas.

The disadvantages of using a different gas is that this is limited to collectors which are air tight. However there may be other reasons, such as reducing condensation for making the collector air tight, in which case the extra cost would only be that of the gas.

One field of interest recently developed in connection with nuclear power stations [8] is that of a mixture of a vapour and a gas to totally suppress convection: \( \text{Nu} = 1 \). In a similar manner to that of solar ponds where no convection occurs due to the salt density gradient suppressing the
convection, so a heavier vapour gradient can also suppress convection. This only happens under certain conditions depending on the temperature, pressure and relative molecular weights of the gas and the vapour. Freon and air, and Carbon tetrachloride and air have been suggested [8]. There may be cause to use this mechanism in a solar collector although the properties are not well enough understood. There are however apparent problems which may arise in using say a layer of carbon tetrachloride in the base of a collector.

(i) The emissivity of the liquid will make it difficult to produce a selective absorber, although this might be overcome by using a suspension of low emissivity particles.
(ii) The liquid will evaporate and then condense on the cover creating an additional heat transfer path via latent heat.

7.1.7 Pressure

Figure 7.14 shows the reduction in $h_C$ with pressure. There are three distinct regions.

Region A Reduction in convection. The Rayleigh number (equation 7.11) is dependent on the pressure of the gas through the density term.

$$\rho = \frac{pM}{RT}$$  \hspace{1cm} (7.19)

So as the pressure decreases the convection decreases. When $Ra = Ra_C$ there is no convection. $Ra = Ra_C$ at the critical pressure $p_C$, which is dependent on the
separation between the cover and absorber

\[ P_c = \frac{42.75}{s^{3/2}} \]

for air, \( T = 373.15 \, \text{k} \)
and \( \Delta T = 180^\circ \text{C} \) \hspace{1cm} (7.20)

**Region B: Conduction only**

When \( R_a < R_{ac} \) there is no convection only conduction so \( \text{Nu} = 1 \) and \( h_c = \frac{k}{s} \). The conductivity of a gas remains constant until the mean free path length \( \lambda \) (see Section 7.1.1), is greater than the plate separation \( s \).

**Region C: Reduction in conduction**

When \( \lambda > s \) the conductivity of the gas is reduced with reducing pressure, the reduction is dependent on the plate temperature, separation and physical condition (see equation 7.7).

**7.1.8 Two cover systems**

It is possible to reduce convective heat losses by inserting extra cover plates, as in Figure 7.15. This allows a lower Rayleigh number for the same gap spacing \( (s) \) from the upper cover to the absorber. The reduction in heat loss on inserting a cover half way between the upper cover and absorber is shown in Figure 7.16. The minima and maxima in heat transfer, for the double cover collector occur at larger overall gap spacing \( (s) \) than for the single cover system.
This is because convection now takes place over smaller gaps and over smaller individual temperature differences. For a given overall s, there is a reduction of more than 50% in convective heat loss with the double cover system. As with a single cover system, heat losses can be reduced further if the gap spacing is increased sufficiently. However, it must be remembered that gap spacing cannot be increased indiscriminately because of increased costs and shading effects. It is also possible to further reduce the heat loss by inserting even more cover plates, but this does have the disadvantage of reducing the energy transmitted because of the energy reflected and absorbed by the covers - see Figure 7.17. With glass covers there is little advantage in having more than two covers. However with high transmission thin film plastics it may be beneficial to have more than two.

7.1.9 Honeycombs as convection suppression devices (C.S.D)

If instead of inserting inner covers to reduce convection as suggested in the previous section, the covers are cut into lengths and placed perpendicular to the absorber plate rather than parallel to it (Figure 7.18) an interesting result emerges. Whereas the reflections of solar radiation at the covers are away from the absorber plate in the parallel case, they are toward the absorber plate in the perpendicular case, and hence they are not lost. Moreover because the partitions can be made quite thin as they no longer have to span large distances, absorption in the partitions can be
avoided with the net result that the solar transmittance of the honeycomb structure can be quite high, at angles of incidence near to the perpendicular [9]. Provided the honeycomb is properly sized the vertical partitions can suppress free convection. If they are of material which is opaque to long wave radiation they can also substantially reduce radiant losses from the absorber plate because of the number of structures the emitted infrared radiation needs to pass through, thereby creating a selective absorber. An emissivity of 0.4 has been measured for a collector with a honeycomb structure ($L/D = 10$) and a non-selective absorber [9]. If on the other hand the honeycomb is transparent to long-wave radiation a selective surface can be used in the absorber plate; the partitions are then used purely for convection suppression.

The honeycomb is seen to divide the air layer between the absorber plate and the glass cover into a number of cells. The question arises: how small must these cells be in order to suppress the convection currents? The question is of some importance since it turns out that if the cells are made too large they can actually augment the free convective heat transfer across the air layer, by breaking up the insulative boundary layers which ordinarily form in the air layer at high $\theta$.

The cell size can be characterized by a hydraulic diameter defined as $D_h = 4 \times \text{(cell cross-sectional area)} \div \text{(cell}$
Ra and Nu will both be based upon the distance across the air layer, L, corresponding to the distance from the absorber plate to the glass cover. The aspect ration \( A \) in this case will refer to the ratio of \( L \) to \( Dh \):

\[
A = \frac{L}{Dh}
\]  (7.21)

For a square cell \( Dh \) is the length of a side. The insertion of a honeycomb into a horizontal collector completely suppresses convection currents for all Ra less than a critical value denoted by \( Ra_c \). In the case of an inclined layer (where \( \theta \) is the angle from the horizontal), it is known that a finite convection motion called the base flow (see Section 7.1.2) exists for any Ra greater than zero. This base flow which is driven by the component of gravity along the heated surface of the honeycomb panel, is expected to make only a modest contribution to the heat transfer, at least until very high Ra numbers are reached. A relationship for \( Ra_c \) was discovered experimentally by Holland \[10\] for a square-celled honeycomb in a horizontal panel.

\[
Ra_c = \frac{a^2 + 3.99^2}{a^2} \]  (7.22)

where \( a_1 = 5\sqrt{\pi} A \)

and \( a = a_1 \) for perfectly conducting side walls

and \( a = 0.75a_1 \) for adiabatic side walls

So for \( Ra < Ra_c \) \( Nu = 1 \)

and for \( Ra > Ra_c \)
\[ \text{Nu} = 1 + 0.0585 \left( 1 - \exp \left\{ -1.19 \frac{(Ra^{1/3} - Ra_c^{1/3})}{Ra_c^{1/3}} \right\} \right) \quad (7.23) \]

for \( \theta = 0^\circ \).

These equations correlate with experimental data to 10%.

Cane et al [11] later went on to examine the correlation for tilted honeycombs.

For \( \frac{Ra}{A^4} < 6000 \); \( 30^\circ < \theta < 90^\circ \); \( A > 4 \)

\[ \text{Nu} = 1 + 0.89 \cos (\theta - 60^\circ) \left( \frac{Ra}{2420A^4} \right)^{2.88} - 1.64 \sin \theta \quad (7.24) \]

This equation fits all data in its range to within \( \pm 7.5\% \).

(Provided differences of the order of 20\% are acceptable it also holds for \( A = 3 \).) For the range \( 0^\circ < \theta < 30^\circ \), linear interpolation between the above equation and the correlation equation 7.23 for \( \theta = 0^\circ \) is recommended. Figure 7.19 compares the \( h_c \) for a collector with a honeycomb to one without.

Work has also been done on hexagonal honeycombs (Figure 7.18) compared to square celled honeycombs. Interestingly the hexagonal honeycomb permits less heat transfer at \( \theta = 0 \) and more for \( \theta > 0 \), presumably because its shape is a closer match to cell structure, see Plate 7.1.

Other configurations of slats (see Figure 7.19) have been looked at [12] none of which are as good as the square celled honeycomb for small angles of tilt.
Insertion of a honeycomb introduces an additional mode of heat transfer across the air layer: namely conduction along the honeycomb partitions from one boundary plate to the other. Often it is possible to design the thickness of the partitions to be so small that this mode of heat loss is insignificant. A heat transfer coefficient due to this mode of heat transfer can be closely estimated from Fourier's Law [9]. This yields

$$h_{ch} = 0.2 \frac{k_w}{D_h L}$$

(7.25)

where $k_w$ and $\delta$ are the thermal conductivity and thickness of the partitions respectively.

For $D_h = 0.01\ m$, $L = 0.05\ m$, $\delta = 0.000076\ m$ and $k_w = 0.193\ \text{W m}^{-1}\ \text{°C}^{-1}$ i.e. square celled honeycomb made of polycarbonate with an aspect ratio of 5, $h_{ch} = 0.00586\ \text{W m}^{-2}\ \text{°C}^{-1}$. Thus the conduction along the honeycomb partitions is negligible.

Polycarbonate appears to be the best material for honeycombs, as it has a relatively high softening temperature and is relatively cheap; costs of £ 5 per m$^2$ (1976) [9] have been estimated for polycarbonate honeycombs.

**Effects of gases other than air in honeycombs**

As yet, nobody has investigated the possible reductions in heat loss that can be attributed to combining the effect of a honeycomb structure with a gas filling other than air. But
assuming that the honeycomb equations with air apply equally to other gases, then the important parameters which dominate the heat loss through the gas are the thermal conductivity and Rayleigh number (see Figure 7.20), the best gas being that which has least thermal conductivity and lowest Rayleigh number.

Provided $Ra < Ra_c$ the heat loss will be dependent on just the conductivity of the gas. For $A = 5, \theta = 0^\circ$ $Ra_c \approx 10^6$. Figure 7.21 shows that for argon and air $Ra$ is never greater than the critical value of $Ra, 10^6$. So the heat transfer of a honeycomb filled with air or argon is dependent on just the thermal conductivity of the gas. Since the thermal conductivity of argon is below air (see Figure 7.1) the heat loss of a honeycomb collector with argon will be less than that of one with air.

7.1.10 Conclusion

The heat transfer coefficient for convection and conduction between the cover and absorber of a flat plate collector is affected by the cover to absorber spacing, tilt angle, gas filling, pressure, number of covers and honeycombs. Figure 7.22 shows the radiative heat transfer coefficient at different absorber temperatures for several possible collector configurations. For comparison Figure 7.22 also shows the relative heat transfer coefficient - there is
little advantage in decreasing the convective and conductive heat transfer coefficient much below the radiative coefficient.

The heat transfer due to convection and conduction can, in theory, be reduced below that of radiation and so substantially improve the performance of solar collectors. The majority of the theory presented in this chapter is also highly relevant to the reduction of heat loss through building glazing.

All of the above mentioned measures for reducing the heat transfer coefficient are not achieved without some form of trade off either in cost or on one of the other parameters affecting the collector efficiency. The substitution of different gases involves producing a hermetically sealed unit plus the expense of the gas fill. Inserting an extra cover reduces the light transmission and increases the cost. Honeycombs involve a cost increase and a reduction in transmission, particularly at large angles of incidence. Increasing the plate to cover separation increases the overshading and material cost. In evacuated collectors the cover and absorber must structurally support atmospheric pressure. This means that glass as a cover material is not feasible as it is prone to cracking, and plastics are generally not as transparent and tend to degrade with UV. The cover plate as well as absorbers need to be thicker, increasing the cost and heat capacity. Also,
structural supports lead to an increase in the thermal conductivity to the cover, $h_s$. Unduly increasing the tilt of collectors (to reduce convection) reduces the incident energy.

Whether the benefits outweigh the disbenefits depends on the rest of the system's performance and costs. However, as a consequence of the work reported in this chapter, the following general recommendations can be made for flat plate collector construction

(i) The cover to absorber separation ($s$) should be greater than or equal to 5cm.

(ii) Xenon provides the largest reduction in $h_c$ for collectors at atmospheric pressure but at present it is too expensive. Argon is the best gas to use because of its substantial reduction in $h_c$ for little extra cost.

(iii) For partially evacuated collectors where only convection is reduced the use of xenon can substantially reduce $h_c$ while at an acceptable pressure of $10^2$Pa. The cost of such small quantities of gas is not considered prohibitive.

(iv) Honeycombs can eliminate heat loss by convection. The use of low conductivity gases such as argon with
honeycombs can also reduce the heat loss by conduction. By constructing the honeycomb from a material which absorbs infrared radiation a honeycomb can also reduce the radiative heat loss. A Polycarbonate honeycomb with an aspect ratio of 5 provides effective improvement to collector performance.

(v) A substantial decrease in \( h_C \) occurs for collector tilt angles 40° - 80°.

(vi) Reducing the conductivity of air by hard evacuation (<10 Pa) is not recommended because of the substantial problems in maintaining such a low pressure over long periods, and the increased collector weight required to prevent the collector collapsing. Also the added benefit in reducing \( h_C \) below that of pure conduction at atmospheric pressure which can be achieved with honeycombs is small because radiation then becomes the dominant heat loss.

(vii) The potential for reducing heat convection by introducing an inverted density gradient by mixing a gas with a vapour requires more investigation.

Although there are many experimental data relating to the heat transfer coefficients between parallel plates, there has
been little experimental work to measure these coefficients in circumstances similar to those experienced in a collector, that is at typical collector plate spacings, temperatures and pressures. The following section outlines a method of measuring the heat transfer under similar conditions to that experienced in a collector while also using relatively simple equipment. It is envisaged that such equipment will be used in the future to verify the behaviour of honeycombs with gas fillings, mixtures of heavy and light gases, mixtures of gases and vapours and novel CSD's. Results are presented on air, argon, Freon and structured polycarbonate at different plate temperatures.
7.2 Experimentation

The equations for the heat transfer coefficients explained in the previous sections are only approximate and have been derived only over a limited range of conditions, so have not been verified over all the values for which they have been assumed to be correct. Generally they have only been tested for air with pressure being the variable for Ra. For these reasons the values obtained in theory may not correspond to those obtained in practice, particularly for gases other than air, although it may be assumed that these equations provide a basis for design. The importance of the variation with temperature has not been investigated. Nearly all previous measurements have been carried out with $\Delta T = 5^\circ C$ which is not very typical of collector operation.

To obtain experimental values for the total heat transfer from the absorber to the cover, that includes convection, conduction, radiation, and any structural conductivity, and to obtain relationships for these variables in cases not yet examined i.e. gas mixtures and gases in honeycombs, a guarded thermal conductivity measurement apparatus has been constructed.

As the values of $h$ to be measured are small, $< 3.0 \text{ Wm}^{-2}\text{C}^{-1}$, and the changes small, $< 0.1 \text{ Wm}^{-2}\text{C}^{-1}$, the apparatus must be accurate. The method used is a variation of the guarded hot plate and wherever possible we have tried to stay within BS
The virtue of this method is that it is an absolute method capable of good precision, provided that a great deal of care is taken in both construction and operation.

The principle is simple. A heater plate is split into two portions, a central area and a surrounding annulus as shown in Figure 7.23. The gap between them is bridged only by such part of the heater former as is necessary to provide sufficient physical support for the whole. The faces of these plates are embedded with thermocouples. The heaters are sandwiched between the sample for which the heat transfer coefficient \( h_C \) is to be measured and a material of similar and known conductivity. The whole lot is then sandwiched between the two water cooled cold plates and lagged with insulation. In this experiment there is a third heater consisting of a wire around the edge of the guard heater, acting as a secondary guard ring heater.

In operation the cold plates and the central heater are maintained at the required temperatures and the guard heaters are adjusted so that the guard and centre thermocouples indicate equality of temperature at equilibrium. Under these conditions the heat flux from the centre heater is assumed to be normal to the plane of the heater and specimens, and therefore there is no gain from, or heat loss to, the guard rings.
The heat transfer coefficient of the panel is calculated from

$$h_p = \frac{q}{\Delta T_A} - h_s$$ \hspace{1cm} (7.26)

Readings for equation 7.26 must be taken at complete equilibrium. Due to the very low heat transfer coefficients, equilibrium times of the order of hours are involved. For this reason power supplies used must be stable over long periods. Since temperature differences of the order of 0.05°C between centre and guard heater plates have a significant effect on the measurement, the monitoring of temperatures is of great importance. Cold plate temperatures must also be maintained to within ± 0.1°C. This is done with a thermostatically controlled bath utilizing a freezer unit.

Although measurements for equation 7.26 should be made with the guard ring temperatures equal to the centre temperature, it is tedious to balance them and so it is more convenient to know the effect of a given amount of off-balance on the answer so that an appropriate correction can be applied. To do this, small adjustments to the guard ring energy are made around the point of equilibrium. These readings are then plotted against apparent conductivity. The centre power must be maintained constant throughout and after each guard adjustment the apparatus must be left for equilibrium to be re-established. A typical plot is shown in Figure 7.24 and the correct conductivity is given by the intercept at zero unbalance. Over small temperature differences the plot is
linear. The virtue of this refinement in technique is that, with experience, the slope of the plot indicates whether the apparatus is responding properly or not, and also it is a measure of the efficiency of the plate, a high slope being undesirable since it indicates a high lateral conductivity between centre and guard partitions of the hot plate.

To enable the study of the convection and conduction heat transfer coefficient of enclosed flat plates, an acrylic test panel with a variable gap was used. The measured heat transfer coefficient \( h \) is made up of three components

\[
h = h_c + h_s + h_r
\]  

(7.27)

where \( h_r \) is due to radiation, \( h_s \) is due to the structure (side supports) and \( h_c \) due to convection and conduction. Since the experimental equipment has been designed principally to investigate \( h_c \) then \( h_s \) and \( h_r \) must be kept to a minimum. This was achieved by constructing a test panel out of acrylic so that \( h_s \) was at a minimum, and by placing aluminium sheets on the inside of the panel (see Figure 7.25). The emissivity of the aluminium sheets was measured so that \( h_r \) could be measured for any temperature using the following equation

\[
h_r = \left( \frac{\alpha}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \right) (T_1^2 + T_2^2) (T_1 + T_2) \]  

(7.28)

where \( \varepsilon_1 \) and \( \varepsilon_2 \) are the emissivities of the surfaces at temperatures of \( T_1 \) and \( T_2 \).
To analyse the effects of pressure on $h_c$ and to obtain $h_c$ for gas mixtures the panel is connected up via a calcium sulphate dryer to a two stage vacuum pump (diffusion and rotary). The problems inherent in this experimental set up are:

(i) the long time required for the system to reach equilibrium;

(ii) the local Nusselt number cannot be measured. Only an average Nusselt number over an area equal to the central heater;

(iii) the temperature difference across the three heaters cannot be measured accurately enough to reduce errors below $\pm 5\%$;

(iv) the panel must be constructed of rather thick material to withstand low internal pressures. This puts up its structural conductivity. At low pressures the panel also bends slightly so some areas of the panel are in better thermal contact than others.
7.2.1 Description of guarded hot plate apparatus

A photograph of the guarded hot plate apparatus is shown in Plate 7.2, and Figures 7.26 is a schematic diagram of the same.

The heater is made of a main central heater (124mm x 124mm, resistance 558Ω at 21°C) which was connected to a stabilised power supply and four guard ring heaters connected in series (total resistance 256Ω at 21°C). The guard and main heaters were made of graphite paper with copper braid contacts attached by silver loaded epoxy.

Two cold plates were manufactured from copper sheets (40cm x 40cm x 3mm). To keep the temperature of the plates constant copper pipes were brazed to the back of the plates, and the water circulated was temperature stabilized by a Grants temperature controlled bath - see Figure 7.27.

The standard insulation used during tests with the test panel was cut from thick cellular expanded polystyrene (Styrofoam SP, thermal conductivity 0.027 Wm⁻¹K⁻¹ at 10°C, manufactured by Dow). A double layer of this insulation was also used to surround the apparatus and so insulate the heater plates and cold plates from room temperature which in turn was controlled by an air conditioning system. To check the calibration of the system two 5cm thick glass fibre samples (k = 0.026 Wm⁻¹K⁻¹) which had been calibrated at the National
Physical Laboratory (NPL) [14] were inserted between the heater plates and cold plates.

Temperature measurements were made with ten chromel alumel thermocouples connected to a Comark digital thermometer 5000. Heat transfer coefficients were measured inside a test panel - see Figure 7.25, made of 1 cm thick acrylic ($K = 0.2 \text{ Wm}^{-2}\text{k}^{-1}$) with 1 mm thick aluminium sheets placed inside. The emissivity of the aluminium sheets was measured as $0.06 \pm 0.005$ by a D+S emissometer.

7.2.2 Method

Equation 7.26 enables the calculation of the heat transfer coefficient $h_p$ across the complete test panel in Figure 7.26. However, if we want to measure $h_c$ we must correct for the heat transfer across the panel walls, and the radiation heat transfer across the panel. The following method outlines how this is done. Also a method for correcting for unbalance in the guard ring temperature is also presented, which could lead to faster experimentation.

If $T_i = T_g$ then heat flow is in the vertical direction.

Under these conditions, in equilibrium

$$\frac{Q}{A} = h_p (T_i - T_o) + h_s (T_i - T_o) \quad (7.29)$$

where $h_p$ is the heat transfer across the panel from the main heater to the cold plate, and $h_s$ is the heat transfer across
the standard insulation. Note that \( h_s \) can be measured by replacing the panel with a piece of insulation identical to the standard insulation. Equation 7.29 then becomes

\[
\frac{Q}{A} = 2h_s (T_1 - T_0)
\] (7.30)

Clearly by measuring \( Q \) (from the electrical supply to the main heater) and by measuring \( T_1 \) and \( T_0 \) in equilibrium \( h_s \) can be found, and so \( h_p \) can be found. The coefficient \( h_p \) consists of three terms if the panel walls are of identical thickness.

\[
\frac{1}{h_p} = \frac{1}{h_r + h_c} + \frac{2}{h_b}
\] (7.31)

\( h_p, h_r \) and \( h_c \) are functions of temperature while \( h_b \) and \( h_s \) are considered to be independent of temperature over the temperature range that measurements are made.

So

\[
h_b = \frac{k}{t}
\] (7.32)

Where \( k \) is thermal conductivity of panel wall. \( h_b \) can therefore be calculated from the textbook value of \( k \) or measured by just inserting the panel's top into the apparatus so that \( h_s = h_b \). Let's now consider \( h_r \).

\[
h_r = \frac{\sigma \left[ \frac{T_1^2 + T_2^2}{2} \right] \left[ T_1 + T_2 \right]}{\left[ \frac{2}{\epsilon} \right] - 1}
\] (7.33)

if the top and bottom radiation shields have the same emissivity \( \epsilon \).

The internal temperatures are related to the external
temperatures of the panel by the following equation,

\[(T_1 - T_2) = (T_1 - T_0) = \frac{Q_p}{A h_b}\]  

(7.34)

given that

\[\frac{Q_p}{A} = h_p (T_1 - T_0)\]  

(7.35)

and \(h_b\) has been calculated from Equation 7.32. Combining equations 7.34 and 7.35 means that the temperature difference across which radiation conduction and convection take place can be calculated. \(h_r\) can then be calculated from equation 7.33 once the emissivity of the radiation shield has been measured. This can be done with an emissometer. The conduction and convection heat transfer coefficient can be calculated from equation 7.31 given that \(h_p\), \(h_b\) and \(h_r\) are now known.

\[h_c = \left[ \frac{1}{\frac{1}{h_p} - \frac{2}{h_b}} \right] - h_r\]  

(7.36)

The value of \(h_c\) calculated is specific for a horizontal parallel plate system of separation (s), with temperature difference \(\Delta T = (T_2 - T_1)\), because all these variables are expected by theory to affect \(h_c\) - see Section 7.1.

The previous analysis applies only if the heat flow is in the vertical direction i.e. \(T_i = T_g\), and when the system is in equilibrium. To obtain this condition may take a long period. "The ASTM method of tests requires that during the
5-hour equilibrium period the temperature difference between the test area and guard ring should not exceed 0.75 per cent of the temperature difference between hot and cold plates [15]. How does $T_i \neq T_g$ effect the measured value of $h_c$?

Let $T_g$ be the mean of the temperatures measured by the 7 thermocouples embedded in the guard ring heater (see Figure 7.26), $T_i$ the mean temperature of the centre heater, and $\Delta \theta = (T_i - T_g)$. When the guard ring is at a higher temperature than the test area, that is $\Delta \theta < 0$, the test area receives heat from the guard section and hence a smaller power input $Q_{exp}$ to the test area from the main heater is required to maintain a given temperature gradient through the specimen. This means that the measured heat transfer coefficient $h_{exp}$ is less than the true heat transfer coefficient $h$, since for $\Delta \theta \neq 0$,

$$h_{exp} = \frac{Q_{exp}}{\Delta \theta} \text{ where } \theta = (T_i - T_0) \quad (7.37)$$

and for $\Delta \theta = 0$,

$$h = \frac{Q}{\Delta \theta} \quad (7.38)$$

The reverse holds for $\Delta \theta > 0$, that is when the guard is cooler than the main heater. Thus $h_{exp} > h$.

Figure 7.24 shows measured values for $h_s$ plotted against the unbalance in guard ring heater. The unbalance ($\Delta \theta \times 100/\theta$), is expressed as a percentage of the temperature difference
between the hot and cold plates. As can be seen from this figure, the error is directly proportional to \((\Delta \theta \times 100)/\theta\) over the range covered (4 to -2 per cent). Woodside, W,[16] has looked at larger unbalances and conducted tests with zero guard ring power input and which still gave a linear relationship between \(h\) and \((\Delta \theta \times 100)/\theta\). Before being able to correct for unbalance it is important to understand what effect it has on the measurements. The simplest case to examine is the effect an unbalance has if the conductivity \((k)\) of a specimen of material is measured, where \(k\) is independent of \(T\). Woodside [16] has calculated that the fractional error in the thermal conductivity is

\[
\frac{\Delta k}{k} = \frac{L}{A} \frac{\Delta \theta}{\theta} \left(\frac{q_o}{k + c}\right)
\]

(7.39)

Where \(q_o\) and \(c\) are constants dependent on the design of the heater. \(q_o\) is the lateral heat transfer directly across the gap. For materials of high conductivity \(q_o\) becomes less significant. \(c\) depends on the size of the heater and the gap width, the larger the gap width between the heaters the smaller the value of \(c\), because less heat goes from the central heater to the cold plate opposite to the guard ring heater. However, a large gap makes it difficult to accurately determine the area \((A)\) over which the central heater is heating. Equation 7.39 shows that the error in the conductivity for a given unbalance, \(\Delta \theta\), is dependent on the conductivity of the material, the larger error occurring for materials of smaller conductivity. Thus it is more important to have a smaller unbalance when testing evacuated panels.
than with non-evacuated panels.

Because each adjustment to the heater takes the equipment 5 hours to stabilise thermally, trying to reach a condition of balance between the guard and central heater temperature can be very time consuming. It is therefore proposed to take three readings with slight unbalance and then to use equation 7.39 to correct for this, with at least one reading to be taken with a positive unbalance and one with a negative unbalance. There is however, one slight problem in this method in that equation 7.39 is true when applied to a solid material specimen where the heat loss is purely by conduction and so the heat transfer coefficient is independent of temperature. But for a test panel the heat transfer coefficient is a function of temperature, so equation 7.39 will only hold for small temperature unbalances.

Errors

The following errors are associated in making a single reading of heat transfer coefficient.

- Heater current ± 0.07%
- Heater voltage ± 0.18%
- Heater area ± 1.13%
- Temperature differences across sample ± 1.33%
- Sample thickness ± 0.25%

The total error is therefore ± 2%. This was checked by
measuring the thermal conductivity of a glass fibre sample which has been calibrated at the National Physical Laboratory and the measurement was found to be within experimental error.

The error in measuring the convective heat transfer across the test panel is ± 5%: there is a ± 2% error in the standard insulation (which is the other side of the heater); there is also an error in the radiation transfer coefficient equation 7.33, and the error in $h_b$ - see equation 7.32.

7.2.3 Results

Table 7.2 shows the results obtained with the thermal conductivity test rig using an acrylic panel with aluminium radiation shields. The heat transfer coefficients were measured with the panel filled with air at several pressures and for various temperatures and also with the panel filled with carbon tetrachloride and Freon R12 ($\text{CCl}_2\text{F}_2$). These results are plotted in Figure 7.28. Also plotted are the theoretical curves for air. The experimental points for air at atmospheric pressure all lie on the theoretical curve within experimental error, except at very small $\Delta T$. This would have little effect on collector operation because when the absorber temperatures is close to ambient the transmissivity and absorptivity dominate the collectors performance. So this effect could go unnoticed in collector operation.
The results for air at pressures of 71-82 torr tend to approach the point at which theory predicts the onset of convection and so agree with theory. However, for air at 0.35 torr there should be no convection and only conduction. Since the conductivity of air is constant over such a small temperature change one would expect $h_c$ also to be constant. However, the experimental results do vary with temperature.

Inserting a thin layer of carbon tetrachloride appears to offer no advantages to that of air. Freon was tried, and the improvement was only slight. There was however no method of verifying that the collector was filled.

Experiments were performed with argon - see Figure 7.29. It was then noticed that a leak into the test panel had developed, due to the stress of vacuum testing. This leak was detectable by the change in thermal conductivity with time. Measurements were then suspended with the acrylic test panel.

The thermal conductivity of structured polycarbonate was then measured at atmospheric pressure. This is a material which is increasingly being used for collector glazing [17] because of its low thermal conductivity, light weight, immense strength and cheapness. The material behaves as a slatted convection suppression device as well as double glazing. For a temperature difference of 5°C a thermal conductivity across
the complete structure of 5.8 Wm\(^{-2}\)K\(^{-1}\) was measured for 10mm structured double walled polycarbonate (SDP) (see Figure 5.6) manufactured by Rohm GmbH (Germany). Assuming that the structured polycarbonate behaves as a slat suppression device \[18\] the effective emissivity of the collector can be calculated as 0.72. This compares with a value of 0.85 as measured for a single sheet of polycarbonate using an emissometer.

The difference between these two results is due to a portion of the thermal radiation being absorbed by a slat before it reaches the cooler plate. This reduces the temperature gradient in the slats and hence the conduction loss. Also, some re-radiation from the slat is absorbed by the warm plate. The material thus behaves as a selective absorber (provided that the equation for slats is appropriate and that a value of 0.72 holds for the emissivity). Figure 7.30 shows the variation of heat transfer with temperature for different thicknesses of structured polycarbonate. This data should not only prove to be useful for collector design but also for factory and greenhouse design where this material is being used as a transparent roofing material.

Further results were not obtained with the thermal conductivity rig as the summer sunshine meant that priority was given to collector rooftop testing.

Improvements were to have been made to the rig -
thermocouples were to be inserted on the inside plates of the mini solar panel so as to directly measure $T_1$ and $T_2$ and a calibrated fibre glass specimen was to have been used for the standard insulation.

The results obtained however did validate this technique for measuring the heat transfer coefficient in a relatively simple and cheap, if slightly time consuming, manner. New results were also obtained for new materials currently being used for collector development.

Although this technique of measuring heat loss has proved useful, it does not give any information about the heat loss in situ, which may change with time. Heat flux meters have been used to measure heat transfer rates [19]. However, this method interferes with the operation of the collector. There exists potential for a non obtrusive method of measurement whereby the temperature drop across the cover could be monitored to measure the plate to ambient heat transfer. This would provide invaluable evidence as to the effect of wind on the collector, and information on any increase of collector heat loss due to ageing of selective absorbers and any evacuated spaces.
CHAPTER 8: CONCLUSION

The world reserves of fossil fuels are limited, yet the world energy consumption is likely to increase as more people demand a better standard of living. Therefore new sources of energy will be essential for the future. If a long term solution is to be found the energy source should be renewable and environmentally benign. One such source is the direct use of solar energy. First impressions suggest that the UK is not the most appropriate country to use solar energy because it receives less solar energy than most other countries and at present has a glut of fossil fuels. However, this is not the case, because the sunnier countries tend not to have the necessary capital to invest in solar energy, and because UK oil and gas reserves will soon be depleted and because a large proportion of the UK energy demand is for low grade heat - a demand well suited for solar energy.

The greatest potential for solar energy is in the domestic heating market. However, if solar energy is to make a large impact, then interseasonal storage of solar energy is essential to overcome the mismatch between supply and demand. Many systems utilizing the interseasonal storage of solar energy are now operational in climates similar to the UK. However, no such large scale systems have been modelled in detail or built in the UK.
To reduce the cost of interseasonal storage many houses need to share the same store so that its surface to volume ratio is kept to a minimum. There is however little improvement in system efficiency for more than 50 houses sharing the same store. The capital cost per house for an interseasonal solar energy system was found to be £3000 (1980) to £6000 (1980) and the system would consist of 15 to 36 m² of solar collector per house and 50 to 130 m³ of store per house. The capital cost of the system can be broken down as roughly, 1/3 for the store, 1/3 for the collector and 1/3 for the distribution and ancillary equipment. One potential interseasonal heating system using air as the working fluid and pebbles as the storage medium (Prometheus) has been investigated. Prometheus was found to be economic when compared to electricity and gas over a 30 year lifetime. However, the economics were found to be critically dependent on the social discount rate and the fuel inflation rate. Prometheus was also analyzed for its production of energy compared to the energy required to construct it, a nett energy loss was discovered for the first 25 years of operation. Suggesting that as much care should be taken in the energy costs of constructing such a device as in the money costs. For systems utilizing stores with temperatures greater than 50°C, containment and insulation of the storage material is required, making the store expensive. In order to reduce the size and so the cost of such stores, the use of high performance collectors is essential. The use of the ground as a cheap store for temperatures below 50°C is a very
attractive option however little work on this very site specific method of storage has been done in the UK.

The size of an interseasonal store required to heat a house can be reduced by insulating the house and so reducing the energy demand. It is currently more cost effective to add insulation to a standard British house than to add a solar heating system. Therefore solar heating systems should only be installed on well insulated houses.

The biggest barriers to utilizing interseasonal storage of solar energy in the UK are not likely to be technical and economic but more to do with the infrastructure of the existing energy supply technologies and the novelty of district heating.

In the UK there are many applications for solar energy other than heating domestic hot water in the late spring, summer and early autumn. However, the majority of collectors tested and in operation are standard water cooled flat plate collectors principally designed for domestic water heating. This has meant that air heating collectors and high performance collectors have had little research and development carried out on their performance in the UK, and yet these types of collector could play a substantial role in meeting the UK energy demand.

The operation of collectors in the UK can be substantially
different to that in the USA, because of the difference in climatic conditions, principally the greater predominance of diffuse and intermittent radiation and the increased wind speeds in the UK. This particularly applies to certain types of air heating collectors and high performance collectors where their performance is more dependent on wind speed and absorber temperature, and there is a greater variation in collector time constants. This means that the standard test methods adopted in the UK (which are similar to those of the USA) test the operation of the collector for USA type climatic conditions of steady state high level insolation, and do not therefore represent the conditions under which solar collectors will operate in the UK. Also, because the climatic conditions for standard tests are not those normally experienced in the UK outdoors, testing outdoors in the UK is very time consuming. So instead of testing collectors under steady state high insolation, a method is proposed to test collectors outdoors under transient conditions combined with zero radiation testing indoors. This reduces the time and cost of testing as neither a solar simulator nor continuous sunshine are needed, while allowing easy replication of results comparable to those obtained under steady state testing, and also results typical of UK operation. This method should in theory be as applicable to testing air as water heating collectors though in practice air collectors present more problems. Single values of $F_{RU_L}$ and $F_{R\tau\alpha}$ are not adequate to fully represent collector operation in the UK given the different types of collectors now available. It is
therefore suggested that $F_{RU_L}$ and $F_{R\tau_a}$ should be measured for various absorber temperatures, wind speeds and mass flow rates. Also the time constant with and without fluid flow should be measured. All these additional parameters can substantially affect the operation of collectors in the UK.

There are many methods of improving the performance of collectors. Reducing the top loss can substantially increase the solar energy collected annually in the UK. A spacing of 5 cm between absorber and cover is recommended to keep top loss to a minimum as in the use of residual gas such as argon and xenon; convection suppression devices and concentrating collectors. However, there is at present no standard method of assessing the relative merits of these various collector types over the period of a year in the UK.
EPILOGUE

This thesis has barely touched on the problem of assessing the role solar energy can play in providing energy for the UK, and many questions lie unanswered. However, research in the UK on solar energy has almost ground to a halt, the UK spending 1/5th the amount on solar thermal research and development per capita that France does. This due to the rape of the North Sea. However, this enjoyment will be short lived and the UK will have to examine the role renewables can play. We will, however, not be able to turn so easily to the technology the Americans have developed to utilize solar energy satisfactorily as we have in the case of nuclear energy and the pressurized water reactor, because the solar source and the demands are different in the two countries, whereas reactors operate the same come rain or shine. However, the nuclear industry has been able to maintain its technological knowhow even though nuclear power is not at present required.

"the CEGB considers it is important to have a PWR technology in the UK as a positive basis for future orders when required!"

Sizewell 'B' PWR Nuclear Power Station, CEGB, 1982.
'it was important that the industry (nuclear) should have faith in the government's commitment to nuclear power. To this end he proposed a programme of constructing 15 GW of nuclear power over the period from 1982 as a basis for the industry's planning.'

The Rt.Hon David Howell MP
Secretary of State for Energy,
Cabinet Minutes, 23 October 1979.

Why is it that such an argument has not worked with solar energy? Does the answer lie in the absence of a solar bomb?