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

Thesis

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Nomenclature

Chapter 2

$A_C$ Collector area ($m^2$)
$A_S$ Storage tank surface area ($m^2$)
$c$ Appropriate specific heat ($J/kg \cdot °C$)
$c_p$ Volume heat capacity at constant pressure ($J/kg \cdot °C$)
$C_h$ Initial capital expenditure per house (£)
$E_T$ Total (accumulated sum) of the radiation falling over a time period of one month on an inclined surface which is above the threshold radiation ($J/m^2$)
$f$ Differential fuel inflation
$F_h$ Fuel cost per year per house (£)
$F_R$ Collector/heat-exchanger efficiency factor
$F'$ Collector efficiency factor
$i$ Discount rate
$I_{th}$ Threshold solar irradiance ($W/m^2$)
$K_h$ Repeated capital expenditure per house (£)
$L$ Monthly total heating demand for space heating and hot water ($J$)
$L_s$ Energy lost from storage tank during the month ($J$)
$MC$ Storage heat capacity ($J/°C$)
$N$ Lifetime of hardware (years)
$n$ Number of years
$P_{Vac}$ Present value cost per house
$Q$ Heat energy ($J$)
$Q_N$ Net heat transferred to storage during the month ($J$)
$Q_T$ Solar energy collected during the month ($J$)
$R_h$ Running costs per year per house (£)
$s$ Pebble shape factor
$T_a$ Ambient temperature ($°C$)
$T_{at}$ Ambient temperature averaged over periods when the radiation level is above the threshold ($°C$)
$T_g$ Monthly average ground temperature ($°C$)
$T_s$ Store temperature ($°C$)
$\bar{T}_s$ Monthly average store temperature ($°C$)
$T_{so}$ Store temperature at the beginning of the month ($°C$)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta T$</td>
<td>Temperature change ($^\circ$C)</td>
</tr>
<tr>
<td>$t_m$</td>
<td>Total number of seconds in a month</td>
</tr>
<tr>
<td>$t_t$</td>
<td>Total number of seconds collector is in operation in month, i.e. when radiation level is above threshold</td>
</tr>
<tr>
<td>$U_L$</td>
<td>Collector overall loss coefficient (W m$^{-2}$ °C$^{-1}$)</td>
</tr>
<tr>
<td>$U_S$</td>
<td>Storage tank heat loss coefficient (W m$^{-2}$ °C$^{-1}$)</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume (m$^3$)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density (kg m$^{-3}$)</td>
</tr>
<tr>
<td>$(\tau \alpha)$</td>
<td>Monthly average transmittance-absorptance product</td>
</tr>
</tbody>
</table>
Nomenclature

Chapter 3

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_C$</td>
<td>Collector area ($m^2$)</td>
</tr>
<tr>
<td>$F_R$</td>
<td>Collector heat-exchanger efficiency factor</td>
</tr>
<tr>
<td>$f$</td>
<td>Fraction of monthly total demand met by solar energy</td>
</tr>
<tr>
<td>$H_T$</td>
<td>Monthly average daily radiation incident on the collector surface per unit area ($Jm^{-2}$)</td>
</tr>
<tr>
<td>$L$</td>
<td>Monthly total heating demand for space heating and hot water ($J$)</td>
</tr>
<tr>
<td>$N$</td>
<td>Days in month</td>
</tr>
<tr>
<td>$T_a$</td>
<td>Monthly average ambient temperature ($^\circ C$)</td>
</tr>
<tr>
<td>$T_{ref}$</td>
<td>An empirically derived reference temperature ($100^\circ C$)</td>
</tr>
<tr>
<td>$t_m$</td>
<td>Total number of seconds in a month</td>
</tr>
<tr>
<td>$U_L$</td>
<td>Collector overall loss coefficient ($Wm^{-2} °C^{-1}$)</td>
</tr>
<tr>
<td>$(\tau \alpha)$</td>
<td>Monthly average transmittance-absorptance product</td>
</tr>
</tbody>
</table>
Nomenclature

Chapter 4

A  Aperture area, or transparent frontal area of collector (m²)

Cp  Specific heat of transfer fluid at constant pressure (J·kg⁻¹·°C⁻¹)

Dh  Characteristic length (m)

F'  Absorber plate (or collector) efficiency factor

FR  Collector heat removal factor

g  Acceleration of gravity (ms⁻²)

h₁  Convective heat transfer coefficient, duct top to heat transfer fluid (W·m⁻²·°C⁻¹)

h₂  Convective heat transfer coefficient, duct base to heat transfer fluid (W·m⁻²·°C⁻¹)

hr  Radiative heat transfer coefficient (W·m⁻²·°C⁻¹)

hw  Wind heat transfer coefficient (W·m⁻²·°C⁻¹)

H  Duct height (m)

I  Equivalent normal solar irradiance (W·m⁻²)

k  Thermal conductivity (W·m⁻¹·°C⁻¹)

L  Collector length (m)

m  Mass flow rate of transfer fluid (Kg·s⁻¹)

Nu  Nusselt number

Pr  Prandtl number

Qₚ  Energy per unit time, useful (W)

Ra  Rayleigh number

Re  Reynolds number

T₁  Duct top, temperature (°C)

T₂  Duct base, temperature (°C)

Ta  Ambient air-temperature (°C)

Tc  Cover temperature (°C)

Te  Exit fluid temperature (°C)

Ti  Inlet fluid temperature (°C)

Tm  Mean fluid temperature (Te + Ti)/2 (°C)

Tp  Average absorber temperature (°C)

Ub  Bottom loss heat transfer coefficient (W·m⁻²·°C⁻¹)

Uₑ  Edge loss heat transfer coefficient (W·m⁻²·°C⁻¹)

UL  Collector overall heat transfer (loss) coefficient (W·m⁻²·°C⁻¹)
$U_t$  Top loss heat transfer coefficient ($\text{Wm}^{-2} \cdot \text{C}^{-1}$)
$V$  Wind velocity ($\text{ms}^{-1}$)
$W$  Collector width (m)
$x$  Insulation thickness (m)
$\alpha$  Absorptance of the collector absorber surface for solar radiation
$\beta$  Volume thermal expansion coefficient ($\text{K}^{-1}$)
$\epsilon_c$  Cover emissivity
$\epsilon_p$  Absorber plate emissivity
$\eta$  Efficiency
$\mu$  Absolute (dynamic) coefficient of viscosity ($\text{Kg m}^{-1} \text{s}^{-1}$)
$\rho$  Density ($\text{Kgm}^{-3}$)
$\tau$  Transmittance of the solar collector
$(\tau\alpha)$  The product of the absorptance of the collector plate and the transmittance of the cover for normal irradiance
$\sigma$  Stefan-Boltzmann constant
Nomenclature

Chapter 5

A  Aperture area, or transparent frontal area for collector (m²)

A_c  Collector area (m²)

c_p  Volume heat capacity at constant pressure (J Kg⁻¹ °C⁻¹)

P'  Absorber plate (or collector) efficiency factor

P''  Collector flow factor

P_1  Correction factor for partial shading of the collector

P_2  Correction factor for variation of τα with the angle of incidence

P_3  Correction factor for variation in optical properties from normal for diffuse irradiance

P_R  Collector heat removal factor

h_w  Wind heat transfer coefficient (W m⁻² °C⁻¹)

I  Equivalent normal solar irradiance (W m⁻²)

I_b  Direct solar irradiance in plane of collector (W m⁻²)

I_d  Diffuse solar irradiance in plane of collector (W m⁻²)

I_m  Measured total solar irradiation incident upon the aperture plane of the collector (W m⁻²)

m  Mass flow rate of transfer fluid (Kg s⁻¹)

m_l  Mass flow rate of leak (Kg s⁻¹)

M  Fluid capacity of collector (Kg)

(m_c)_e  Effective heat capacity of collector (J °C⁻¹)

q  Output power per unit aperture area conveyed by the heat transfer fluid (W m⁻²)

Q_u  Energy per unit time, useful (W)

(Q_u)_t  Energy per unit time under transient conditions (W)

r  Correlation coefficient

t  Time (s)

T_a  Ambient air temperature (°C)

T_b  Average back plate temperature (°C)

T_e  Exit fluid temperature (°C)

T_f  Average temperature of the fluid in the collector (°C)

T_i  Inlet fluid temperature (°C)
Measured fluid inlet temperature (°C)
Mean fluid temperature \((T_e + T_i)/2\) (°C)
Absorber plate temperature (°C)
Mean absorber temperature (°C)
Equivalent black body sky temperature (°C)
Reduced temperature \((T_i - T_a)/I\) (m² °C w⁻¹)
Collector overall heat transfer (loss) coefficient (Wm⁻² °C⁻¹)
Wind velocity (ms⁻¹)
Efficiency
Product of the absorptance of the collector plate and the transmittance of the cover for normal irradiance.
Collector time constant under flow conditions (s)
Cut off time (s)
Effective transmittance absorptance product
Product of the absorptance and transmittance for normal irradiance
Time increment
Angle of incidence; degrees from normal
Nomenclature
Chapter 6

$F_R$ Collector heat removal factor

$h_{p-c}$ Convection coefficient between absorber plate and cover ($Wm^{-2}°C^{-1}$)

$h_{r-c-a}$ Radiation coefficient between absorber plate and cover ($Wm^{-2}°C^{-1}$)

$h_{r-c-a}$ Radiation coefficient from the cover to sky ($Wm^{-2}°C^{-1}$)

$h_w$ Wind heat transfer coefficient. ($Wm^{-2}°C^{-1}$)

$I$ Equivalent normal solar irradiance ($Wm^{-2}$)

$I_{th}$ Threshold solar irradiance ($Wm^{-2}$)

$T_a$ Ambient air temperature ($°C$)

$T_i$ Inlet fluid temperature ($°C$)

$U$ Collector heat loss coefficient $P'U_L$ ($Wm^{-2}°C^{-1}$)

$U_L$ Collector overall heat transfer (loss) coefficient ($Wm^{-2}°C^{-1}$)

$\epsilon_t$ Thermal emissivity

$\eta$ Efficiency steady state

$\eta_d$ Daily averaged efficiency

$\eta_o$ Zero loss collector efficiency, $P'(\alpha \tau)$.

$\tau_s$ Solar transmissivity

$(\tau \alpha)_s$ Product of the absorptance and transmittance for normal irradiance
### Nomenclature

#### Chapter 7

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Aspect ratio or area of main heater</td>
</tr>
<tr>
<td>a</td>
<td>Accommodation coefficient</td>
</tr>
<tr>
<td>c̄</td>
<td>Average velocity of molecules (ms⁻¹)</td>
</tr>
<tr>
<td>( c_p )</td>
<td>Specific heat at constant pressure (J Kg⁻¹ °C⁻¹)</td>
</tr>
<tr>
<td>( c_v )</td>
<td>Specific heat at constant volume (J Kg⁻¹ °C⁻¹)</td>
</tr>
<tr>
<td>d</td>
<td>Molecular diameter (m)</td>
</tr>
<tr>
<td>( D_h )</td>
<td>Hydraulic diameter (m)</td>
</tr>
<tr>
<td>g</td>
<td>Acceleration of gravity (ms⁻²)</td>
</tr>
<tr>
<td>Gr</td>
<td>Grashof number</td>
</tr>
<tr>
<td>h</td>
<td>Combined heat transfer coefficient from absorber to cover (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>( h' )</td>
<td>Heat transfer coefficient of material of known conductivity (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>( h_b )</td>
<td>Heat transfer coefficient for flow across panel wall (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>( h_c )</td>
<td>Heat transfer coefficient for flow across the inside of the panel due to convection and conduction (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>( h_p )</td>
<td>Heat transfer coefficient for flow across panel (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>( h_r )</td>
<td>Heat transfer coefficient for flow across the inside of the panel due to radiation (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>( h_s )</td>
<td>Heat transfer coefficient for flow across standard insulation (Wm⁻² °C⁻¹)</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity (Wm⁻¹ °C⁻¹)</td>
</tr>
<tr>
<td>L</td>
<td>Linear dimension (m)</td>
</tr>
<tr>
<td>m</td>
<td>Wall molecule mass (Kg)</td>
</tr>
<tr>
<td>m'</td>
<td>Gas molecule mass (Kg)</td>
</tr>
<tr>
<td>M</td>
<td>Mass of one mole (kg mol⁻¹)</td>
</tr>
<tr>
<td>( N_A )</td>
<td>Avogadro's number</td>
</tr>
<tr>
<td>Nu</td>
<td>Nussult number</td>
</tr>
<tr>
<td>p</td>
<td>Gas pressure (Nm⁻²)</td>
</tr>
<tr>
<td>( P_c )</td>
<td>Critical pressure when ( R_a = R_{a_c} )</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>q</td>
<td>Power dissipated in central heater (W)</td>
</tr>
</tbody>
</table>
\( Q \)  
Energy per unit time, rate of heat supply to main heater (W)

\( Q_p \)  
Rate of heat supply to panel from main heater (W)

\( r \)  
Specific gas constant (R/M)

\( R \)  
Gas constant

\( Ra \)  
Rayleigh number

\( Ra_C \)  
Critical Rayleigh number, for \( Ra < Ra_C \) no convection, \( Nu = 1 \)

\( Re \)  
Reynolds number

\( s \)  
Absorber plate to cover separation (m)

\( t \)  
Panel wall thickness (m)

\( T \)  
Average of plate and cover temperature (°C)

\( T_1 \)  
Inside panel temperature nearest to cold plate (°C)

\( T_2 \)  
Inside panel temperature nearest to main heater (°C)

\( T_g \)  
Guard ring temperature (°C)

\( T_i \)  
Temperature of main heater, also fluid inlet temperature (°C)

\( T_0 \)  
Temperature of cold plates (°C)

\( \alpha \)  
Thermal diffusivity (m² s⁻¹)

\( \beta \)  
Thermal volume expansion coefficient (= 1/T for a perfect gas), (K⁻¹)

\( \gamma \)  
\( \frac{cp}{cv} \)

\( \Delta \theta \)  
Hot plate temperature unbalance (\( T_i - T_g \)), (°C)

\( \Delta T \)  
Temperature difference across panel (°C)

\( \epsilon_1 \)  
Emissivity of surface at temperature \( T_1 \) (°C)

\( \epsilon_2 \)  
Emissivity of surface at temperature \( T_2 \) (°C)

\( \mu \)  
Viscosity (Pa s)

\( \nu \)  
Kinematic viscosity (\( \mu/\rho \)) (Pa s m³Kg⁻¹)

\( \rho \)  
Density (Kg m⁻³)

\( \sigma \)  
Stefan-Boltzmann constant (Wm⁻² K⁻⁴)

\( \lambda \)  
Mean free path (m)
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<td>Specific heat capacity $C_P$/JK$^{-1}$ K$^{-1} \times 10^3$</td>
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<td>2.1</td>
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<td>Water and salt</td>
<td>(brine)</td>
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<td>Alumina (Al$_2$O$_3$)</td>
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<td>4.0</td>
<td>0.9</td>
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<td>0.963</td>
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<td>Cracking occurs at high temperature</td>
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<tr>
<td>Caloria HT43 (oil)</td>
<td>Cracking occurs at high temperature</td>
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<td></td>
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<td>Cracking occurs at high temp.</td>
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<td>Zero voids (30% void $\rho C_P = 1.7$)</td>
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<td>Stone</td>
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<td>Material</td>
<td>Density</td>
<td>Strength</td>
<td>Cost (1980) £25/m³</td>
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<tr>
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<td>Dry earth</td>
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### TABLE 2.3 Basic Prometheus configuration to heat 100 houses

**Store**

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<td>width</td>
<td>10 m</td>
</tr>
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<td>height</td>
<td>4 m</td>
</tr>
<tr>
<td>volume</td>
<td>11200 m³</td>
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<td>storage material pebbles, density</td>
<td>1600 kg m⁻³</td>
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<td>storage material pebbles; specific heat capacity</td>
<td>837 J kg⁻¹ °C⁻¹</td>
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<td>store insulation; thickness</td>
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<td>store insulation; thermal conductivity</td>
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**Collector**

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<td>heat transfer factor (F_R)</td>
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<td>overall heat loss coefficient</td>
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<td>optical efficiency averaged over useful incident angles (τα)</td>
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| TOTAL & (1978) (1980) | 458607 | | | | | |
TABLE 2.5  Present value of the costs per house of 3 space and water heating systems, $N = 45$ years, $n_1 = 15$ years, $n_2 = 30$ years. Domestic space and water heating requirement = 27.5 $G J/yr$, costs in £ 1980.

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<td>$K_h/£$</td>
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<td>$F_h/£ \text{yr}^{-1}$</td>
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<td>$R_h/£ \text{yr}^{-1}$</td>
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$i=0.05 \ f=0.04$ \ 6600 \ 6000 \ 6300

$PVC_h$ $i=0 \ f=0.04$ \ 8500 \ 17800 \ 20200

$i=0 \ f=0.02$ \ 7500 \ 11700 \ 12500
### TABLE 2.6 Costs and inventory of various interseasonal solar heating systems modelled along with the cost, collector area and storage volume required to provide 27.5 GJ per annum.

<table>
<thead>
<tr>
<th>System Description</th>
<th>P.C.L. [11]</th>
<th>ERR [11]</th>
<th>Studsvik [67]</th>
<th>Prometheus</th>
</tr>
</thead>
<tbody>
<tr>
<td>Collector type</td>
<td>Flat plate selective</td>
<td>Evacuated tube collector</td>
<td>Concentrating collector</td>
<td>High performance evacuated</td>
</tr>
<tr>
<td>Collector area /m²</td>
<td>2100</td>
<td>4600</td>
<td>14000</td>
<td>2800</td>
</tr>
<tr>
<td>Storage volume /m³</td>
<td>7500</td>
<td>17700</td>
<td>38500</td>
<td>11200</td>
</tr>
<tr>
<td>Insulation thickness/m</td>
<td>1.0</td>
<td>0.4</td>
<td>0.3</td>
<td>0.6</td>
</tr>
<tr>
<td>Operating temperature of store/°C</td>
<td>72-42</td>
<td>95-60</td>
<td>70-30</td>
<td>130-30</td>
</tr>
<tr>
<td>Number of houses heated by system</td>
<td>50</td>
<td>300</td>
<td>400</td>
<td>100</td>
</tr>
<tr>
<td>Energy consumption GJ/annum per house</td>
<td>32.4</td>
<td>25</td>
<td>54</td>
<td>27.5</td>
</tr>
<tr>
<td>Cost of collectors £_1980/m²</td>
<td>60</td>
<td>64</td>
<td>64</td>
<td>72</td>
</tr>
<tr>
<td>Cost of store £_1980/m³</td>
<td>16</td>
<td>11</td>
<td>10</td>
<td>26</td>
</tr>
<tr>
<td>Collector area/Storage volume (m²/m³)</td>
<td>0.28</td>
<td>0.26</td>
<td>0.36</td>
<td>0.25</td>
</tr>
<tr>
<td>Total system capital cost £_1980</td>
<td>322900</td>
<td>659000</td>
<td>1740000</td>
<td>570000</td>
</tr>
<tr>
<td>Collector area required to heat type A5 house (27.5 GJ/annum)/m²</td>
<td>35.7</td>
<td>16.9</td>
<td>17.8</td>
<td>28</td>
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<tr>
<td>Storage volume required for type A5 house /m³</td>
<td>127</td>
<td>65</td>
<td>49</td>
<td>112</td>
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<tr>
<td>Cost per A5 house/£_1980</td>
<td>5480</td>
<td>2416</td>
<td>2215</td>
<td>5700</td>
</tr>
</tbody>
</table>

[ ] Chapter 2 reference numbers
<table>
<thead>
<tr>
<th>Store temperature rise/(°C)</th>
<th>Cost/£1982 per KWh recovered energy seasonal storage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel tank</td>
<td>80</td>
</tr>
<tr>
<td>Pit storage</td>
<td>50</td>
</tr>
<tr>
<td>Rock cavern</td>
<td>70</td>
</tr>
<tr>
<td>Storage in clay</td>
<td>12</td>
</tr>
<tr>
<td>Multiple well systems in rock</td>
<td>50</td>
</tr>
<tr>
<td>Aquifers</td>
<td>15</td>
</tr>
<tr>
<td>Prometheus (pebble bed, using data from Table 2.6)</td>
<td>100</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Name</th>
<th>Location of Store/or Centre of Study</th>
<th>Design Study or Constructed</th>
<th>Storage Material</th>
<th>Number of Houses Per Store</th>
<th>% of Annual House Heating Supplied by System</th>
<th>Cost Per House £</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lambohov, Sweden</td>
<td>Constructed</td>
<td>Water</td>
<td>56</td>
<td>100</td>
<td></td>
<td>27 000</td>
</tr>
<tr>
<td>Inglestad, Sweden</td>
<td>Constructed</td>
<td>Water</td>
<td>52</td>
<td>50</td>
<td></td>
<td>19 320</td>
</tr>
<tr>
<td>Studsvik, Sweden</td>
<td>Design Study</td>
<td>Water</td>
<td>400</td>
<td>93</td>
<td></td>
<td>5 150</td>
</tr>
<tr>
<td>Lyckebo, Sweden</td>
<td>Design Study</td>
<td>Water</td>
<td>500</td>
<td>100</td>
<td></td>
<td>10 500</td>
</tr>
<tr>
<td>Arizona, USA</td>
<td>Design Study</td>
<td>Water</td>
<td>250</td>
<td>100</td>
<td></td>
<td>3 012</td>
</tr>
<tr>
<td>Northampton, USA</td>
<td>Design Study</td>
<td>Solar Pond</td>
<td>10 000</td>
<td>100</td>
<td></td>
<td>6 000</td>
</tr>
<tr>
<td>Sussex, UK</td>
<td>Design Study</td>
<td>Solar Pond</td>
<td>100</td>
<td>100</td>
<td></td>
<td>10 000</td>
</tr>
<tr>
<td>City University,</td>
<td>Design Study</td>
<td>Water</td>
<td>100</td>
<td>78</td>
<td></td>
<td>4 000</td>
</tr>
<tr>
<td>London, UK</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ERR, UK</td>
<td>Design Study</td>
<td>Water</td>
<td>300</td>
<td>100</td>
<td></td>
<td>2 416</td>
</tr>
<tr>
<td>PCL, UK</td>
<td>Design Study</td>
<td>Water</td>
<td>50</td>
<td>100</td>
<td></td>
<td>5 480</td>
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</table>
TABLE 3.1 Thermal Characteristics of Basic Type AO House

<table>
<thead>
<tr>
<th>Component</th>
<th>Variable</th>
<th>Area A (m²)</th>
<th>U-value (Wm⁻²°C⁻¹)</th>
<th>UA (W°C⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall</td>
<td></td>
<td>88.5</td>
<td>1.0</td>
<td>88.5</td>
</tr>
<tr>
<td>Roof</td>
<td></td>
<td>48.6</td>
<td>0.6</td>
<td>29.2</td>
</tr>
<tr>
<td>Floor</td>
<td></td>
<td>48.6</td>
<td>0.5</td>
<td>24.3</td>
</tr>
<tr>
<td>Window</td>
<td></td>
<td>15.0</td>
<td>5.5</td>
<td>82.5</td>
</tr>
<tr>
<td>Total fabric specific loss</td>
<td></td>
<td></td>
<td></td>
<td>224W°C⁻¹</td>
</tr>
<tr>
<td>Ventilation specific loss</td>
<td></td>
<td></td>
<td></td>
<td>80W°C⁻¹</td>
</tr>
<tr>
<td>Total house specific loss</td>
<td></td>
<td></td>
<td></td>
<td>304W°C⁻¹</td>
</tr>
</tbody>
</table>
### Average weather data (1969-1977) for Kew, London, Latitude 51°N

<table>
<thead>
<tr>
<th>Month</th>
<th>Days in month</th>
<th>Solar radiation on a South-facing vertical surface (KWh/m²/month)</th>
<th>Solar radiation on a South-facing surface 30° to horizontal (KWh/m²/month)</th>
<th>Ambient Temperature (°C)</th>
<th>Degree days baseline 15.5°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jan</td>
<td>31</td>
<td>28</td>
<td>25.2</td>
<td>5.2</td>
<td>346</td>
</tr>
<tr>
<td>Feb</td>
<td>28</td>
<td>42</td>
<td>45</td>
<td>4.6</td>
<td>304</td>
</tr>
<tr>
<td>March</td>
<td>31</td>
<td>74</td>
<td>91</td>
<td>5.7</td>
<td>282</td>
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<tr>
<td>April</td>
<td>30</td>
<td>75</td>
<td>115</td>
<td>8.2</td>
<td>197</td>
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<tr>
<td>May</td>
<td>31</td>
<td>87</td>
<td>146</td>
<td>11.8</td>
<td>113</td>
</tr>
<tr>
<td>June</td>
<td>30</td>
<td>90</td>
<td>166</td>
<td>14.9</td>
<td>-</td>
</tr>
<tr>
<td>July</td>
<td>31</td>
<td>84</td>
<td>150</td>
<td>17.2</td>
<td>-</td>
</tr>
<tr>
<td>Aug</td>
<td>31</td>
<td>78</td>
<td>123</td>
<td>16.8</td>
<td>-</td>
</tr>
<tr>
<td>Sept</td>
<td>30</td>
<td>72</td>
<td>95</td>
<td>13.9</td>
<td>56</td>
</tr>
<tr>
<td>Oct</td>
<td>31</td>
<td>59</td>
<td>66</td>
<td>10.8</td>
<td>132</td>
</tr>
<tr>
<td>Nov</td>
<td>30</td>
<td>39</td>
<td>37</td>
<td>6.7</td>
<td>256</td>
</tr>
<tr>
<td>Dec</td>
<td>31</td>
<td>25</td>
<td>22</td>
<td>5.3</td>
<td>333</td>
</tr>
<tr>
<td>House type</td>
<td>Insulation level</td>
<td>Total house specific loss ($W^O C^{-1}$)</td>
<td>Net annual space and water heating demand (GJ)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>------------</td>
<td>------------------</td>
<td>-----------------------------------------</td>
<td>---------------------------------------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A0</td>
<td>Basic (1975 Building Regs.)</td>
<td>304</td>
<td>46.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A1</td>
<td>A0 + orientate house north-south</td>
<td>304</td>
<td>41.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A2</td>
<td>A1 + 50 mm loft insulation (100 mm total)</td>
<td>291</td>
<td>39.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A3</td>
<td>A2 + fill cavity with fibre</td>
<td>255</td>
<td>33.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A4</td>
<td>A3 + 50 mm loft insulation (150 mm total)</td>
<td>251</td>
<td>33.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A5</td>
<td>A4 + extra layer of glazing (i.e. double)</td>
<td>213</td>
<td>27.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A6</td>
<td>A5 + cavity increased to 100 mm</td>
<td>186</td>
<td>23.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A7</td>
<td>A6 + 25 mm floor edge insulation</td>
<td>182</td>
<td>22.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A8</td>
<td>A7 + all windows on south side</td>
<td>182</td>
<td>20.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A9</td>
<td>A8 + 100 mm of loft insulation (250 mm total)</td>
<td>177</td>
<td>19.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A10</td>
<td>A9 + extra layer of glazing (i.e. triple)</td>
<td>164</td>
<td>18.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A11</td>
<td>A10 + cavity increased to 200 mm</td>
<td>150</td>
<td>16.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Component</td>
<td>Area A (m²)</td>
<td>U-value (Wm⁻²°C⁻¹)</td>
<td>UA (W°C⁻¹)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>----------------------</td>
<td>-------------</td>
<td>--------------------</td>
<td>------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wall</td>
<td>73.9</td>
<td>1.0</td>
<td>73.9</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Roof</td>
<td>41.2</td>
<td>0.6</td>
<td>24.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Floor</td>
<td>41.2</td>
<td>0.5</td>
<td>20.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Window</td>
<td>13.3</td>
<td>5.5</td>
<td>73.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total fabric specific loss</td>
<td></td>
<td></td>
<td>192 W°C⁻¹</td>
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</tr>
<tr>
<td>Ventilation specific loss</td>
<td></td>
<td></td>
<td>68 W°C⁻¹</td>
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</tr>
<tr>
<td>Total house specific loss</td>
<td></td>
<td></td>
<td>260 W°C⁻¹</td>
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</tr>
</tbody>
</table>
TABLE 3.5  Thermal Characteristics of existing houses with different levels of retrofitted insulation.

<table>
<thead>
<tr>
<th>House type</th>
<th>Insulation level</th>
<th>Total house specific loss ($W^0C^{-1}$)</th>
<th>Net annual space water heating demand (GJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BO</td>
<td>Basic (average UK housing stock)</td>
<td>260</td>
<td>34.9</td>
</tr>
<tr>
<td>B1</td>
<td>BO + 50 mm of loft insulation (100 mm total)</td>
<td>249</td>
<td>33.1</td>
</tr>
<tr>
<td>B2</td>
<td>B1 + fibre-fill cavity (50 mm)</td>
<td>219</td>
<td>28.3</td>
</tr>
<tr>
<td>B3</td>
<td>B2 + 50 mm of loft insulation (150 mm total)</td>
<td>215</td>
<td>27.7</td>
</tr>
<tr>
<td>B4</td>
<td>B3 + extra layer of glazing (i.e. double)</td>
<td>182</td>
<td>23.1</td>
</tr>
<tr>
<td>B5</td>
<td>B4 + extra layer of glazing (i.e. triple)</td>
<td>170</td>
<td>21.7</td>
</tr>
<tr>
<td>B6</td>
<td>B5 + 100 mm external wall insualtion</td>
<td>156</td>
<td>19.6</td>
</tr>
<tr>
<td>Year</td>
<td>Location</td>
<td>Collector Name</td>
<td>Collector Type</td>
</tr>
<tr>
<td>------</td>
<td>----------</td>
<td>----------------</td>
<td>----------------</td>
</tr>
<tr>
<td>1980</td>
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<td>28-15-00</td>
<td>口中口口</td>
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TABLE 4.1: collector, test locations and material systems in the United Kingdom.
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<th>4</th>
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<th>9</th>
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<tr>
<td>25 June 1983</td>
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<tr>
<td>26 June 1983</td>
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<tr>
<td>5 July 1983</td>
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</tbody>
</table>

**Table 5.1** Data collected during steady state testing of the D.C. Hall collector

<table>
<thead>
<tr>
<th>Column</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
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<tbody>
<tr>
<td>19 Aug. 1983</td>
<td>1209</td>
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<td>59.2</td>
<td>83.9</td>
<td>76.4</td>
<td>2.7</td>
<td>27.1</td>
<td>0.05</td>
<td>0.06</td>
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<td>1210</td>
<td>64.0</td>
<td>0.0</td>
<td>27.4</td>
<td>60.2</td>
<td>81.6</td>
<td>76.7</td>
<td>3.3</td>
<td>27.9</td>
<td>0.06</td>
<td>0.08</td>
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<td>27.4</td>
<td>59.2</td>
<td>81.6</td>
<td>76.7</td>
<td>1.8</td>
<td>27.2</td>
<td>0.05</td>
<td>0.07</td>
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<td>0.0</td>
<td>27.3</td>
<td>58.9</td>
<td>81.4</td>
<td>76.5</td>
<td>1.7</td>
<td>25.0</td>
<td>0.05</td>
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**Column Index**

1. Time (hrs. min)
2. Mass flow rate (kg hr⁻¹)
3. Total insolation (W m⁻²)
4. Air temperature rise passing through collector (T₂ - T₁, °C)
5. Ambient air temperature (°C)
6. Inlet air temperature (°C)
7. Outlet air temperature (°C)
8. Absorber temperature (°C)
9. Wind speed (ms⁻¹)
10. Efficiency (n)
11. (T₂ - T₁) / T
12. Absorber temperature - T₂ / T

---

Note: The table provides data collected during steady state testing of the D.C. Hall collector on various dates from June to August 1983. The columns represent different parameters such as time, mass flow rate, total insolation, air temperature rise, ambient air temperature, and outlet air temperature.
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<th>Test No.</th>
<th>Date</th>
<th>Time of test</th>
<th>Air mass flow rate</th>
<th>Air temp. at inlet</th>
<th>Air temp. at outlet</th>
<th>Air temp. increase ($T_e - T_i$)</th>
<th>Ambient Temp.</th>
<th>Total irradiance in plate of collector ($I_m$)</th>
<th>Collector efficiency</th>
<th>Wind speed</th>
<th>Absorber Temp.</th>
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<td>69.9</td>
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<td>18/8/83</td>
<td>1142-1151</td>
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<td>17.1</td>
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<td>&lt;0.4</td>
<td>51.4</td>
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</table>
\[
\begin{array}{cccccccc}
\text{Test No.} & \text{Date} & \text{Day/night, h} & \text{Air mass, kg} & \text{Air temp. at outlet, } (T_e-T) & \text{Air temp. increase, } (T_e-T) & \text{Total collector area, } m^2 & \text{Airflow, } m^3/\text{hr} & \text{Wind speed, } m/s \\
1 & 8/6/83 & 125-130 & 83.5 & 19/6/83 & 128-132 & 71.9 & 21.3 & 47.7 & 26.8 \\
2 & 8/6/83 & 125-130 & 83.5 & 21/6/83 & 120-123 & 71.6 & 24.6 & 43.5 & 26.8 \\
3 & 19/6/83 & 130-134 & 72.1 & 7/6/83 & 117-120 & 77.2 & 21.9 & 44.5 & 30.5 \\
4 & 7/6/83 & 117-120 & 77.2 & 6/6/83 & 110-113 & 77.9 & 21.3 & 47.7 & 25.8 \\
5 & 16/6/83 & 115-118 & 80.4 & 16/6/83 & 115-118 & 80.4 & 21.3 & 47.7 & 25.8 \\
6 & 159-150 & 75.8 & 26.0 & 159-150 & 75.8 & 26.0 & 21.3 & 47.7 & 25.8 \\
7 & 7.2 & 14.4 & 26.0 & 7.2 & 14.4 & 26.0 & 21.3 & 47.7 & 25.8 \\
8 & 5.6 & 14.4 & 26.0 & 5.6 & 14.4 & 26.0 & 21.3 & 47.7 & 25.8 \\
9 & 4.5 & 14.4 & 26.0 & 4.5 & 14.4 & 26.0 & 21.3 & 47.7 & 25.8 \\
10 & 4.5 & 14.4 & 26.0 & 4.5 & 14.4 & 26.0 & 21.3 & 47.7 & 25.8 \\
11 & 4.5 & 14.4 & 26.0 & 4.5 & 14.4 & 26.0 & 21.3 & 47.7 & 25.8 \\
12 & 4.5 & 14.4 & 26.0 & 4.5 & 14.4 & 26.0 & 21.3 & 47.7 & 25.8 \\
\end{array}
\]

TABLE 5.2(b) Results of steady state testing of structured polycarbonate collector
TABLE 5.3 Collector configuration modelled for transient analysis by RRDCT.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tr>
<td>Collector length (along flow)</td>
<td>4.00 m</td>
</tr>
<tr>
<td>Collector width</td>
<td>1.00 m</td>
</tr>
<tr>
<td>Cover to plate spacing</td>
<td>0.05 m</td>
</tr>
<tr>
<td>Rear Duct gap</td>
<td>0.01 m</td>
</tr>
<tr>
<td>Back insulation dry glass fibre</td>
<td>0.10 m</td>
</tr>
<tr>
<td>Edge insulation dry glass fibre</td>
<td>0.05 m</td>
</tr>
<tr>
<td>Material of plate and duct-back</td>
<td>duraluminium HS 15 TB</td>
</tr>
<tr>
<td>Plate absorbtance</td>
<td>0.95 at $\theta = 0$ falling slightly as $\theta$ increases</td>
</tr>
<tr>
<td>Emissivity of upper surface of the plate (diffuse)</td>
<td>0.10</td>
</tr>
<tr>
<td>Emissivity of duct surface (diffuse)</td>
<td>0.91</td>
</tr>
<tr>
<td>Emissivity of cover (diffuse)</td>
<td>0.85</td>
</tr>
<tr>
<td>Cover polycarbonate thickness</td>
<td>2.00 mm</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>0.06 kg s$^{-1}$</td>
</tr>
<tr>
<td>Thickness of plate and of duct-back</td>
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</tr>
<tr>
<td>Dy1</td>
<td>0.2 mm</td>
</tr>
<tr>
<td>Dy2</td>
<td>0.5 mm</td>
</tr>
<tr>
<td>Dy3</td>
<td>1.0 mm</td>
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<tr>
<td>Dy4</td>
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<tr>
<td>Dy5</td>
<td>5.0 mm</td>
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TABLE 5.4 Results of transient and steady state testing with multi node model

<table>
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<tr>
<th>Parameter</th>
<th>Steady state</th>
<th>Transient 0.5mm (DY2)</th>
<th>Transient 2mm (DY4)</th>
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<tr>
<td>$\Delta t$/min</td>
<td>-</td>
<td>1</td>
<td>4</td>
</tr>
<tr>
<td>N</td>
<td>-</td>
<td>6</td>
<td>5</td>
</tr>
<tr>
<td>$\tau_c$/min</td>
<td>-</td>
<td>2.8</td>
<td>9.7</td>
</tr>
<tr>
<td>$F_{RUL}/(\text{Wm}^{-2}\text{K}^{-1})$</td>
<td>2.83*</td>
<td>2.768</td>
<td>2.604</td>
</tr>
<tr>
<td>$K_{R\tau}$</td>
<td>0.683</td>
<td>0.585</td>
<td>0.569</td>
</tr>
<tr>
<td>$K_F R\tau$</td>
<td>0.683</td>
<td>0.706</td>
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<tr>
<td>$\hat{\sigma} F_{RUL}$</td>
<td>-</td>
<td>0.012</td>
<td>0.036</td>
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<tr>
<td>$\hat{\sigma} F_{R\tau}$</td>
<td>-</td>
<td>0.0008</td>
<td>0.0025</td>
</tr>
</tbody>
</table>

$K = \text{correction factor for equivalent normal direct radiation} = \frac{(r\alpha)_{\text{direct}}}{(r\alpha)_{\text{diffuse}}} = 0.830 \div 0.688 = 1.206$

$* = \text{at low fluid inlet temperatures}$
<table>
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<th>Column 1</th>
<th>Column 2</th>
<th>Column 3</th>
<th>Column 4</th>
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<td>76</td>
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**Table 5.5**

Transient test data for Sp collector input to fans

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<th>1. Data number</th>
<th>2. Incident insolation in (W/m²)</th>
<th>3. Output power per unit aperture area, g (W/m²)</th>
<th>4. (i - iₐ)</th>
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<tr>
<td>TABLE 5.6 Data Output from 'TRANS' for SP collector, n = 1, in the format specified in Table F.6.1 of British Standard DD 77: 1982</td>
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<td>---</td>
<td>---</td>
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</tr>
<tr>
<td>n</td>
<td>( F_R(t_a), k_n )</td>
<td>( dF_R(t_a), k_n )</td>
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### TABLE 5.8
Temperature distribution within DY1 collector (0.2mm thick plate and duct back) during ASHRAE steady state testing, $T_a = 293k$, $I = 700\text{w}m^{-2}$, $\text{Wind} = 1\text{m s}^{-1}$, $T_{sky} = 273k$

<table>
<thead>
<tr>
<th>$T_i/k$</th>
<th>$T_e/k$</th>
<th>$\overline{T_p}/k$</th>
<th>$\overline{T_b}/k$</th>
<th>$T_m/k$</th>
<th>$F\text{RUL}_{\text{U}}$ ($Wm^{-2}^\circ C^{-1}$)</th>
<th>$\eta$</th>
<th>$F_{\text{ave}}\text{U}_{\text{L}}$ ($Wm^{-2}^\circ C^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>303</td>
<td>332.73</td>
<td>333.01</td>
<td>322.1</td>
<td>317.86</td>
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### TABLE 5.9
Temperature distribution and energy lost from DY1 collector (0.2mm thick plate and duct base) during zero radiation testing, $T_a = 293k$, $T_{wind} = 1\text{m s}^{-1}$, $T_{sky} = 273k$

<table>
<thead>
<tr>
<th>$T_i/k$</th>
<th>$T_e/k$</th>
<th>$\overline{T_p}/k$</th>
<th>$\overline{T_b}/k$</th>
<th>$T_m/k$</th>
<th>Energy lost per unit time per unit area $Wm^{-2}$</th>
<th>$F\text{RUL}_{\text{U}}$ ($Wm^{-2}^\circ C^{-1}$)</th>
<th>$F_{\text{ave}}\text{U}_{\text{L}}$ ($Wm^{-2}^\circ C^{-1}$)</th>
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<td>303</td>
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<td>300.41</td>
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<td>366.41</td>
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<td>374.20</td>
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<td>398.43</td>
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<td>406.34</td>
<td>412.12</td>
<td>418.73</td>
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<td>301.71</td>
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* $T_{sky} = 293k$
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<th>28</th>
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<th>Indoor</th>
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<tr>
<td>400° &gt; 362</td>
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<td>5.4</td>
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<td>Transient</td>
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<tr>
<td>400° &gt; 326</td>
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<td>4.2</td>
<td>0.557</td>
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<td>400° &gt; 274</td>
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<td>3.1</td>
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Structured Polymercondate Collector

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<tr>
<th></th>
<th>600</th>
<th>2</th>
<th>20</th>
<th>2.46</th>
<th>Theory</th>
</tr>
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<tr>
<td>300° &gt; 225</td>
<td>225</td>
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<td>-</td>
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<td>0 &gt;= Radiaton</td>
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<tr>
<td>300° &gt; 200</td>
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<td>0.583</td>
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<tr>
<td>700° &gt; 175</td>
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<td>0.627</td>
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D.C. Hall Collector

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<th></th>
<th>(mm)/Im</th>
<th>(mm)/Im</th>
<th>C0/4C</th>
<th>C0/4C</th>
<th>P/uL</th>
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Test method

Summary of collector testing results

TABLE 5.10
<table>
<thead>
<tr>
<th>Material</th>
<th>Reflective index (n)</th>
<th>Solar Transmittance (0.2-4.0μm)</th>
<th>Infrared Transmittance (3.0-500μm)</th>
<th>Expansion Coefficient (°C^-1)</th>
<th>Temperature Limits (°C)</th>
<th>Weather-ability (comments)</th>
<th>Chemical Resistance (comments)</th>
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<tbody>
<tr>
<td>Lexan (Polycarbonate)</td>
<td>1.586</td>
<td>125 mil</td>
<td>125 mil</td>
<td>7.98 x 10^-5</td>
<td>120-130</td>
<td>Good</td>
<td>Good</td>
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<tr>
<td>Plexiglass (Acrylic)</td>
<td>1.49</td>
<td>125 mil</td>
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<td>8.29 x 10^-5</td>
<td>80-90</td>
<td>Average to good</td>
<td>Good to excellent</td>
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<tr>
<td>Teflon F.F.P. (Fluorocarbon)</td>
<td>1.343</td>
<td>5 mil</td>
<td>5 mil</td>
<td>12.55 x 10^-5</td>
<td>200-220</td>
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<td>Excellent</td>
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<tr>
<td>Tedlar P.V.F. (fluorocarbon)</td>
<td>1.46</td>
<td>4 mil</td>
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<td>110-170</td>
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<td>Excellent</td>
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<tr>
<td>Mylar (Polyester)</td>
<td>1.64-1.67</td>
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<td>5 mil</td>
<td>2.00 x 10^-5</td>
<td>150-200</td>
<td>Poor</td>
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<tr>
<td>Sunlite (Fibre glass)</td>
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<td>25 mil</td>
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<td>2.98 x 10^-5</td>
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<td>125 mil</td>
<td>10.21 x 10^-6</td>
<td>230</td>
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<tr>
<td>Temper glass (Glass)</td>
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<td>125 mil</td>
<td>10.21 x 10^-6</td>
<td>230-250</td>
<td>Excellent</td>
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<td>Clear limesheet glass (Low iron glass)</td>
<td>1.51</td>
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<td>125 mil</td>
<td>10.64 x 10^-6</td>
<td>200</td>
<td>Excellent</td>
<td>Good to excellent</td>
</tr>
<tr>
<td>Clear lime temper glass (Low iron glass)</td>
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<td>125 mil</td>
<td>10.64 x 10^-6</td>
<td>200</td>
<td>Excellent</td>
<td>Good to excellent</td>
</tr>
<tr>
<td>Sunade white crystal glass (0.01% iron glass)</td>
<td>1.50</td>
<td>125 mil</td>
<td>125 mil</td>
<td>10.00 x 10^-6</td>
<td>200</td>
<td>Excellent</td>
<td>Good to excellent</td>
</tr>
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Source: Gary, H.P. 'Treatise on solar energy' Vol.1, A Wiley Interscience Publication, Chichester, 1982
<table>
<thead>
<tr>
<th>Material</th>
<th>Substrate</th>
<th>Supporter</th>
<th>Trade name</th>
<th>Supplier</th>
<th>Location</th>
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<th>Thermal</th>
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<td>Olympic</td>
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<td>TABLE 6.3</td>
<td>Key to collector variable features, used to obtain Figure 6.19</td>
<td></td>
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<td></td>
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</table>

| Cover Material: | Cover 1 | Plate Glass, Thickness 6.0 mm |
| Cover 2 | Polycarbonate, Thickness 2.0 mm |

| Thickness of the Plate and of the Duct-Back: |
| DY1 | 0.2 mm |
| DY2 | 0.5 mm |
| DY3 | 1.0 mm |
| DY4 | 2.0 mm |
| DY5 | 5.0 mm |

| Air Flow in the Rear-Duct: |
| Flow 0 | Stagnation (\( M = 0 \)) |
| Flow 1 | All \( T_I = 303 \) K \( M = 0.0600 \) kg s\(^{-1}\) \( \text{PON} \) irrelevant |
| Flow 2 | \( T_I = 303 \) K \( M = 0.0600 \) kg s\(^{-1}\) \( \text{PON} = 128W \) |
| Flow 3 | \( T_I = 323 \) K \( M = 0.0562 \) kg s\(^{-1}\) \( \text{PON} = 124W \) |
### TABLE 7.1 Some typical thermal accommodation coefficients

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<th>Gas</th>
<th>Surface</th>
<th>Surface condition (absorbed gas)</th>
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<td></td>
<td>W</td>
<td>Clean</td>
<td>-196</td>
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<td>Pt</td>
<td>Saturated</td>
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<tr>
<td></td>
<td>K</td>
<td>Clean</td>
<td>25</td>
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<td>Na</td>
<td>Clean</td>
<td>25</td>
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<td>Kr</td>
<td>W</td>
<td>Clean</td>
<td>30</td>
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<td>W</td>
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<td>-196</td>
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<td></td>
<td>W</td>
<td>Clean</td>
<td>-183</td>
<td>0.942</td>
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TABLE 7.2  Convection and conduction heat transfer coefficients for various gases at different temperatures as measured with guarded hot plate.

<table>
<thead>
<tr>
<th>Condition</th>
<th>$T_s/°C$</th>
<th>$T_i/°C$</th>
<th>$h_p/(\text{Wm}^{-2}°\text{C}^{-1})$</th>
<th>$Q_p/\text{Wm}^{-2}$</th>
<th>$T_r/°C$</th>
<th>$h_r/(\text{Wm}^{-2}°\text{C}^{-1})$</th>
<th>$h_c/(\text{Wm}^{-2}°\text{C}^{-1})$</th>
<th>$\Delta T/°C$</th>
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<tr>
<td>Air at atmospheric pressure</td>
<td>10</td>
<td>14</td>
<td>0.798</td>
<td>3.19</td>
<td>10.16</td>
<td>13.84</td>
<td>0.163</td>
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<td>10.1</td>
<td>20.7</td>
<td>1.910</td>
<td>20.05</td>
<td>11.10</td>
<td>19.70</td>
<td>0.168</td>
<td>2.193</td>
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<td>10.1</td>
<td>21.3</td>
<td>1.725</td>
<td>19.32</td>
<td>11.07</td>
<td>20.33</td>
<td>0.169</td>
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<td>10.5</td>
<td>33.3</td>
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<td>30.80</td>
<td>0.180</td>
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<td>Air, $p = 82$ torr</td>
<td>10.35</td>
<td>37.9</td>
<td>1.60</td>
<td>44.08</td>
<td>12.55</td>
<td>35.70</td>
<td>0.185</td>
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<td>10.35</td>
<td>38.8</td>
<td>1.621</td>
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<td>36.49</td>
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<td>40.44</td>
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<td>0.847</td>
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<td>22.1</td>
<td>1.685</td>
<td>19.88</td>
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<td>10.73</td>
<td>17.17</td>
<td>0.166</td>
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<td>Carbon Tet/Air</td>
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<td>17.9</td>
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<td>10.74</td>
<td>17.26</td>
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<td>13.53</td>
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<td>10.38</td>
<td>16.42</td>
<td>0.165</td>
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<td>Air $p = 0.35$ torr</td>
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<td>12.87</td>
<td>43.72</td>
<td>0.193</td>
<td>1.088</td>
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<td>Air $p = 16$ torr and changing</td>
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<td>51.2</td>
<td>1.186</td>
<td>47.46</td>
<td>13.57</td>
<td>48.83</td>
<td>0.198</td>
<td>1.148</td>
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</table>
FIGURE 1.1(b) HISTOGRAM OF ENERGY CONSUMPTION PER CAPITA FOR DIFFERENT PHYSICAL QUALITY OF LIFE INDEX (PQLI) FOR THE PEOPLE OF THE WORLD. THE PERCENTAGES SHOWN IN EACH BAR ARE THE PERCENTAGES WITHIN THAT RANGE OF PQLI.
FIGURE 2.1 UK LOW GRADE HEAT, FUEL CONSUMPTION AND END USE.

FIGURE 2.2 DOMESTIC SPACE AND HOT WATER DEMAND.
Figure 2.3  Distribution of annual gas consumption for 90 similar houses in Milton Keynes, from 'The performance of domestic wet heating systems', Pickup, G.A.C.7

Figure 2.4  Weekly consumption of hot water for one household, from 'The performance of domestic wet heating systems', Pickup, G.A.C.7
Total No of dwellings 87
Overall mean weekly consumption 0.841 m³/week
Standard deviation 0.351 m³/week

Contribution due to
OAPs flats
(1 or 2 occupants)

Dwelling mean weekly hot water consumption m³

**Figure 2.5** MEAN WEEKLY HOT WATER CONSUMPTION FOR 87 VARIOUS SITES.
FROM 'THE PERFORMANCE OF DOMESTIC WET HEATING SYSTEMS'
BY G.A. RICKUP [9]

**Figure 2.6** SOLAR AND THERMAL RADIATION SPECTRAL DISTRIBUTIONS.
AIR MASS m = 0 IS FOR EXTRA-TERRESTRIAL RADIATION,
m = 2 IS A TYPICAL CITY DISTRIBUTION.
FIGURE 2.7
ANNUAL VARIATION OF MEAN DAILY TOTALS OF DIRECT AND DIFFUSE INSO LATION ON A HORIZONTAL SURFACE.

**FIGURE 2.9** DEMONSTRATION PROJECT IN STUDSVIK. [26]

**FIGURE 2.10**

FIGURE 2.11  SEASONAL HEAT STORAGE AND A CENTRAL SHORT TERM STORAGE RESERVOIR (C.S.T.) CONSTRUCTED FOR TNO DELFT [35]

ONE-FAMILY HOUSES (SMALL SCALE)

with heat storage in preferably soft ground or clay solid rock

APARTMENT BUILDING (INTENSE POPULATED AREAS) (LARGE SCALE)

with heat storage in
- preferably solid rock
- most types of ground

FIGURE 2.12  DIFFERENT APPLICATIONS FOR 'SUNSTORE' [37], SEASONAL STORAGE IN THE GROUND
**FIGURE 2.13** PLAN OF PROMETHEUS RETROFITTED TO SUPPLY 83 HOUSES WITH ALL THEIR SPACE HEATING AND HOT WATER.

**FIGURE 2.14** COLLECTOR MOUNTED ON TOP OF STORE, PART OF PROMETHEUS DESIGN.
PROTOTYPE OF A PROMETHEUS TYPE SOLAR AIR-COLLECTOR/HEAT STORE, INSTALLED AT THE OPEN UNIVERSITY, MILTON KEYNES, UK.
FIGURE 2.16  INSOLATION INCIDENT ON PROTO- PROMETHEUS, 28TH SEPTEMBER 1981

FIGURE 2.17  COLLECTOR, STORE AND AMBIENT TEMPERATURES FOR PROTO- PROMETHEUS ON 28TH SEPTEMBER 1981.
FIGURE 2.19 PROTO-PROMETHEUS TEMPERATURE DISTRIBUTION (WITH FAN ON), ON 22ND SEPTEMBER 1981 AT 14:25 HR.
SAMPLE SIZE 204
AVERAGE 1.6 cm.
STANDARD DEVIATION 0.7 cm.

FIGURE 2.19 FREQUENCY DISTRIBUTION OF PEBBLE SMALLEST DIMENSION.
FIGURE 2.20  FREQUENCY DISTRIBUTION OF PEBBLE LARGEST DIMENSION

SAMPLE SIZE  204
AVERAGE   3.8cm
STANDARD DEVIATION  0.95cm
Figure 2.21 Proto-Prometheus store temperature, from 22nd September 1981 to 2nd October 1981 under stagnation (fan off).

Figure 2.22 Energy demand for a 3-bedroom house built to E75 building regulations (Type A) with solar heating supplied by a basic type Prometheus.
**Figure 2.23**  
Effect of changing the collector overall heat loss coefficient on the % of annual energy supplied by Prometheus on a Type AI house.

**Figure 2.24**  
Effect of changing the collector area on the % of annual energy supplied by Prometheus to a Type AI house.
FIGURE 2.25  THE EFFECT OF CHANGING THE STORAGE TANK INSULATION THICKNESS ON THE % OF SOLAR ENERGY SUPPLIED BY PROMETHEUS TO A TYPE A1 HOUSE.

FIGURE 2.26  THE EFFECT OF CHANGING THE STORAGE VOLUME ON THE % OF SOLAR ENERGY SUPPLIED BY PROMETHEUS TO A TYPE A1 HOUSE.
FIGURE 2.27  THE EFFECT OF INCREASING THE NUMBER OF HOUSES SERVED BY A SINGLE CUBIC PROMETHEUS (SIZE 112 m² PER HOUSE AND 28 m² OF COLLECTOR PER HOUSE) FOR A TYPE A1 HOUSE.

FIGURE 2.28  THE EFFECT OF CHANGING THE COLLECTOR OVERALL HEAT LOSS ON THE % OF ENERGY SUPPLIED BY A CUBIC PROMETHEUS HEATING A TYPE A5 HOUSE.
**Figure 2.29** Design of Costed Prometheus to provide 100% of their annual heating demand (27.5 GJ) with solar energy.

**Figure 2.30** Improved collector orientation
Figure 3.1  Design of basic type AO house

Figure 3.2  Net space heating demand for type AO, A5 and A11 3 bedroom end of terrace house.
FIGURE 3.3
USEFUL ENERGY SAVED AND EXTRA COST FOR VARIOUS INSULATION OPTIONS AND SOLAR SYSTEMS INSTALLED WHILE CONSTRUCTING A BASIC TYPE A0 HOUSE.

FIGURE 3.4
ENERGY DEMAND FOR A 3 BEDROOM TERRACE BUILT TO 1975 BUILDING REGULATIONS AND ENERGY SUPPLIED BY 4, 12 AND 24 m² OF SOLAR COLLECTOR.
**Figure 3.5** Energy Demand for a Well Insulated 3 Bedroom House, and Energy Supplied by 4, 12, and 24 m$^2$ of Solar Collector.

**Figure 3.6** Comparison of Predicted Solar Energy Supply for a House Using the F-Chart Method with the Measured Solar Supply for the Milton Keynes Solar House.
Figure 37: Useful energy saved and extra costs for various insulation options and solar systems retrofitted to an existing type B0 house.
FIGURE 4.1. NONPOROUS ABSORBER-TYPE AIR HEATERS.

FIGURE 4.2. POROUS ABSORBER-TYPE AIR HEATERS.
FIGURE 4.3  HYBRID PHOTOVOLTAIC AND AIR HEATING SOLAR COLLECTOR

FIGURE 4.4  COLLECTOR HEAT LOSSES
**FIGURE 4.5** REAR DUCT COLLECTOR CONFIGURATION

**FIGURE 4.6** TOP DUCT COLLECTOR CONFIGURATION
**Figure 4.7** Correlations for Wind Heat Loss Coefficient

**The Curves Correspond to the Following Relations:**

- **MCDONAGH**
  \[ h_w = 5.7 + 3.5v \]

- **WARMUFF**
  \[ h_w = 2.8 + 3.0v \]

- **LLOYD**
  \[ h_w = 0.15 \times \frac{\dot{Q} \, \alpha}{A \times k} \quad \text{for} \quad T_w = 10^\circ C, T_a = 15^\circ C, L = 1m, W = 1m, \]

- **SPARROW**
  \[ h_w = \frac{k \times 0.96 \times R \, \alpha}{L + W} \quad \text{for} \quad T_w = 10^\circ C, T_a = 15^\circ C, L = 1m, W = 1m \]

- **GREEN**
  \[ h_w = \left( h_{w1} + h_{w2} \right)^{\frac{3}{2}} \quad \text{for} \quad A = 1.4m^2, 45^\circ \text{inclination} \]

- **KIND**
  
  For collector length 2.4m, width 1.2m, height 0.5m, \( T_a = 25^\circ C \)
Figure 4.8 Flow Diagram of 'EFFIC2' (See Appendix B) A Program To Calculate The Efficiency of a Flat-Plate Air Heating Collector.
INPUT
ENVIRONMENTAL PARAMETERS $I, V, T_a$

COLLECTOR CONFIGURATION $(Dx), e, C_f, k, H, A, L, W, D$
$u_e, x$

COLLECTOR VARIABLES $T_c, m$

INITIAL ESTIMATE OF $T_F, T_m$

CALCULATE

$R_C$
$R_C$
$N_w$
$h_1, h_2$
$h_r$
$U_b$
$U_b$
$U_L$
$F_r$
$F_r$
$Q_u$

$\eta = Q_u/AI$

CALCULATE NEW ABSORBER TEMPERATURE

$T_{F_{new}} = T_c + \frac{(Q_u/A)(1-F_r)}{U_L F_r}$

IF $|T_{F_{new}} - T_F| < 0.01$

OUTPUT
$z, T_F, U_L, U_e, U_b, F_r, F_r, Q_u$

FIGURE 4.9 FLOW DIAGRAM OF 'EFFIC' (SEE APPENDIX B) A PROGRAM TO CALCULATE THE EFFICIENCY OF A TYPICAL AMERICAN COLLECTOR.
**Figure 4.10** Response of zero and long time constant collector to changing insulation.
**FIGURE 4.11** NODAL CONFIGURATION OF A FLAT PLATE, REAR-DUCT AIR HEATING, SOLAR COLLECTOR AS USED IN 'RRDCT'.

**FIGURE 4.12** COMPARISON OF AIR OUTLET TEMPERATURE TO PREDICTED BY THE COMPUTER MODEL (SOLID CURVE) AND LABORATORY MEASUREMENTS, ON A SIMILAR, THOUGH NOT IDENTICAL, COLLECTOR (CROSSES).
Figure 4.13  Efficiency curve generated by transient model operating under steady state conditions and steady state model. For collector parameters see Table 5.3.
FIGURE 5.1  PERCENTAGE OF ENERGY FALLING ABOVE A THRESHOLD
INTENSITY AVERAGED OVER A PERIOD OF ONE HOUR
EACH MONTH ON A HORIZONTAL SURFACE (AT NN1966-1975)
SECTION X-X

**Figure 5.2** D.C. Hall Collector
Figure 5.3: Angular variation of transmittance of 2mm thick polycarbonate (Refractive index = 1.586, extinction coefficient $= 0.01\rho\cdot\text{cm}^{-1}$)

Figure 5.4: Tee-pieces used for absorber fins in D.C. Hall collector
FIGURE 5.5-5.6 AIR HEATING COLLECTOR MADE OF STRUCTURED POLYCARBONATE

FIGURE 5.7 SOLAR TRANSMITTANCE OF STRUCTURED POLYCARBONATE VERSUS INCIDENT ANGLE. SOURCE: H.L. REDFOOT ET AL., "GLAZING SOLAR COLLECTORS WITH ACRYLIC AND DOUBLE WALL POLYCARBONATE PLASTICS"
Figure 5.8  Orifice Plate and its Location for Measuring Mass Flow Rate
**Figure 5.9** ASHRAE Standard 93-77 Testing Configuration for a Solar Collector when the Transfer Fluid is Air.

**Figure 5.10** Open University Air Collector Testing Configuration.
**EXPERIMENTAL**

Theoretical \( (T_e = 0.8, T = 0.6, C = 0.9, U = 8 \text{ Wm}^{-2} \text{T}) \)

\[ (hC)_e = 3300 \text{ J/K} \]

**Figure 5.11** Response of structured polycarbonate collector to a step change in insolation from 750 Wm\(^{-2}\) to zero with a fluid flow rate of 72 kg/hr\(^{-1}\).

**Figure 5.12** Uninterrupted insolation as defined by ASHRAE standard 93-77 [2].
FIGURE 5.13 RECORD OF INCIDENT SOLAR RADIATION ON A HORIZONTAL SURFACE AT THE OPEN UNIVERSITY ON 19/6/83.

FIGURE 5.14 RECORD OF INCIDENT SOLAR RADIATION ON A HORIZONTAL SURFACE AND WIND SPEED ON 21/6/83 (CONTINUED ON NEXT PAGE).
Figure 5.15 Angle of incidence of solar radiation onto D.C. Hall collector during steady state efficiency test. Position of collector, Milton Keynes, latitude 52°, longitude 0.78° (horizontal).

Figure 5.16 Angle correction for D.C. Hall collector.
FIGURE 5.17(a) AIR HEATING COLLECTOR UNDER TEST WITH A LEAK AT THE INLET

FIGURE 5.17(b) AIR HEATING COLLECTOR UNDER TEST WITH A LEAK AT THE OUTLET
**Figure 5.18**

The effect of air leaks on the measured value of $F_{\text{UL}}$, for $m = 0.5 \text{ kg/hr}$. 

**Figure 5.19**

Calibration curve for periflow orifice plate for air at 20°C.
Figure 5.20: Pressure distribution within collector test configuration with and without fluid flow.

Figure 5.21: Sample output of DC Hall collector to testing outdoors not under steady state conditions.
**Figure 5.22**  Steady State Efficiency Curve for D.C. Hall Collector Tested Outdoors.

**Figure 5.23**  Steady State Efficiency Curve for Structured Polycarbonate Collector Tested Outdoors.
**Figure 5.24** Uncorrected Efficiency Curve with Variation of Wind Speed between 0 - 4 m/s. Source: [25].

**Figure 5.25** Efficiency Curve Corrected for Variation in Wind Speed Using a Normalizing Function. Source: [25].
FIGURE 5.26 VARIATION OF MASS FLOW RATE CAUSED BY CHANGE IN WIND SPEED
FIGURE 5.27 ROUND ROBIN TESTING OF LIQUID FLAT PLATE COLLECTORS. 
THE COMBINED EFFECT OF METEOROLOGICAL EXTREMES 
AND MEASUREMENT UNCERTAINTY. SOURCE: TAYLOR [28].

FIGURE 5.28 MEASURED DEPENDENCY OF $P(\text{CO}_2)$ ON THE DIFFUSE FRACTION FOR A 
SINGLE-GLAZED FLAT-PLATE COLLECTOR. SOURCE: PORSKI [34].
**Figure 5.29**  
**Computer generated steady state and transient efficiency curve for 0.5 mm absorber plate**
FIGURE 5.30 TRANSIENT DIFFUSE RADIATION

FIGURE 5.31 FLUID OUTLET TEMPERATURE UNDER TRANSIENT CONDITIONS

FIGURE 5.32 INTEGRATED RESPONSE OF COLLECTOR OVER 1 AND 2 MINUTES TO TRANSIENT RADIATION.
FIGURE 5.33
THE VARIATION IN $F_{uL}$, $F_{cL} \theta$, AND $\phi_{cL}$ WITH THE NUMBER OF INCREMENTS USED IN THE TRANSIENT ANALYSIS.
FIGURE 5.34  COLLECTOR RESPONSE FUNCTIONS FOR OPTIMUM VALUES OF N.

FIGURE 5.35  CALCULATED COLLECTOR TIME CONSTANTS, FOR DIFFERENT COLLECTOR CONFIGURATIONS SEE TABLE 5.3.
FIGURE 5.36  EFFICIENCY CURVE GENERATED FROM TRANSIENT TESTING RESULTS OF THE SP COLLECTOR AND PROCESSED BY "TRANS" FOR N=1, UNCORRECTED FOR ANGLE OF INCIDENCE OF RADIATION.

FIGURE 5.37  TRANSIENT INSOLATION DURING TESTING OF SP COLLECTOR ON 17/6/83, CONTINUED ON NEXT PAGE.
Figure 5.37 continued. Transient Insolation during Testing of SP Collector on 14/6/93 - 15/6/93.
**Figures 5.38 and 5.39**

- **Figure 5.38** Standard error in $F_{UL}$ versus $N$ the number of previous time steps influencing the collector's present performance under transient conditions for the structured polycarbonate collector.

- **Figure 5.39** Efficiency curve for outdoor transient testing of structured polycarbonate collector. Data generated from 'TRANS' for $N=7$, uncorrected for angle of incidence of radiation.
**Figure 5.40** Collector response function for S.P. collector N=7.

**Figure 5.41** Efficiency curve for outdoor transient testing of D.C. Hall collector (Manors abbreviated). Data generated from TRANS for N=7, uncorrected for incident angle of radiation.
**Figure 5.42** Indoor Solar Collector Test Facility.

**Figure 5.43** Relative Spectral Intensity of 'Cool Ray' Lamps, Transmittance of Polycarbonate and Reflectance of Maxorb.
FIGURE 5.44  INTENSITY DISTRIBUTION ACROSS COLLECTOR DURING INDOOR TESTING IN W/m², AVERAGE INTENSITY 2.11 W/m², STANDARD DEVIATION ± 0.1 W/m².

FIGURE 5.45  WING GENERATOR.
Figure 5.46 Variation of wind speed (m/s), 5 mm above collector surface

Figure 5.47 Measured and predicted heat loss U* for D.C. wall collector (non-selective) with varying wind speed indoors.
Figure 5.48 Efficiency curve of structured polycarbonate collector measured indoors and outdoors.

Figure 5.49 Efficiency curve of D.C. Hall collector with non-selective absorber (Nestel). Indoor measurements and computer predictions.
FIGURE 5.50 REDESIGNED INDOOR COLLECTOR TEST FACILITY

FIGURE 5.51 STEADY STATE AND ZERO TESTING EFFICIENCY CURVES.
Figure 5.52: Steady state and efficiency curve plotted against mean absorber plate temperature ($T_p$) for simulated collector.
**Figure 5.53**: Steady State and Zero Testing Efficiency Curve Plotted Against Mean Fluid Temperature ($T_m$) for Simulated Collector.
**Figure 5.54** Collector temperature profile for model collector under steady state and zero testing conditions for the same fluid inlet temperature (303K).

**Figure 5.55** Collector temperature profile for model collector under steady state and zero testing conditions for the same mean absorber plate temperature (366K).
**FIGURE 5.56**  Temperature of absorber and rear duct for the same average fluid temperature with the collector under zero and steady state testing.

**FIGURE 5.57**  $F_{W}/U$ versus mean fluid temperature for collector D1 under zero testing and average steady state testing.
FIGURE 5.58 EFFICIENCY CURVES FOR D.C. HALL COLLECTOR USING DIFFERENT TEST METHODS
Figure 5.59 Efficiency Curve for Structured Polycarbonate Collector Under Different Test Conditions

Figure 5.60 Top Loss Coefficient Versus Absorber Temperature for A & D Chaff Type Collector (Glass Absorber)
Figure 5.6: Steady State Efficiency of Solar Collector (Black-Chrome) measured during operation and indoor testing, source: Taylor, P.J. 'Performance of Selective and Non-Selective Solar Thermal Absorbers in a Working Installation,' Solar World Congress edited by S.V. Szomboly, Vol. 2, pp. 1159-1153.
Figure 6.1  Efficiency curve for 'Conventional' and 'High Performance' collector.

Figure 6.2  Typical construction of a flat plate collector.
Figure 6.4  Percentage of energy falling above a threshold intensity averaged over a period of one hour each month on a horizontal surface (April 1978-May 1979).

Figure 6.5  Maximum improvement to flat plate collector performance by increasing $T_1$ and $a$. 
**Figure 6.6** Reflectance of Solar Collector Coatings

**Figure 6.7** Steady State Efficiency of Solar Collector Measured During Operation and Indoor Testing. Source: Taylor, P.J., Performance of Selective and Non-Selecting Solar Thermal Absorbers in a Working Installation, Solar World Congress ed by S.N. Sivolov, vol 2, pp 1149-1153.
Figure 6.8: Efficiency curves for different methods of heat loss reduction.

FIGURE 6.10  EFFICIENCY CURVE OF ADVANCED FLAT PLATE COLLECTOR WITH XENON BETWEEN THE ABSORBER AND COVER AT A PRESSURE OF 1 TORR.

FIGURE 6.11  EFFICIENCY VERSUS MASS FLOW RATE FOR STRUCTURED POLYCARBONATE COLLECTOR. \( I_{in} = 2.1 \text{ Wm}^{-2} \), \( T_a = 28^\circ\text{C} \), \( T_{in} > T_a \), \( T_c = T_a \) and AIR VELOCITY = 1.5 \text{ m} \cdot \text{s}^{-1} \)
FIGURE 6.12  PRESSURE DROP ACROSS S.P. COLLECTOR VERSUS MASS FLOW RATE

FIGURE 6.13  THEORETICAL SYSTEM EFFICIENCY VERSUS MASS FLOW RATE FOR A FLUID INLET TEMPERATURE OF 60°C, FOR THREE PHT SPFATIONS z, AND TWO LEVELS OF INCIDENT INSULATION.
FIGURE 6.14  EFFICIENCY CURVE FOR A COMBINED PARABOLIC CONCENTRATOR COMPARED
WITH A FLAT PLATE COLLECTOR. SOURCE: ARQUONIAT NATIONAL LABORATORY TECH REPORT.

FIGURE 6.15  GLOBAL AND DIFFUSE INSOLATION MONTH BY MONTH AT 45° SOUTH FACED
SLOPE.
**Figure 6.16**
Annual energy collected versus collector temperature. Comparison of five types of collector. Source [33].

**Figure 6.17**
Figure 6.18  Simulated ambient conditions. For further details see text in Appendix C.

WIND = 1.0 m s⁻¹

TK = TA - 20, clear skies
TK = TA - 10, overcast skies
Figure 6.19 Steady-state efficiency ($\eta$ - the solid curve) and daily averaged efficiency ($\bar{\eta}$). The values of $\bar{\eta}$ are for a variety of simulated conditions (see Table I and Figure 4).

(i) S0J/TAJ, flow 2  (ii) SOM/TAM, flow 2  (iii) SOD/TAD1, flow 2  (iv) SOM/TAM, flow 3  (v) SLM/TAM, flow 2  (vi) SOD/TAD1, flow 3  (vii) S1D1/TAD1, flow 2  (viii) SOD/TAD2, flow 3  (ix) S1D2/TAD1, flow 2  (x) S1D3/TAD1, flow 2  (xi) S1D/TAD1, flow 2.
FIGURE 6.20 'FMTC' AIR HEATING SOLAR COLLECTOR DEVELOPED BY GE [42]

Figure 6.22. Instantaneous efficiencies of the PMTC collector and a single glazed flat plate collector and their variation with insolation. [42]
**Figure 7.1** Thermal conductivity of various gases at 20°C versus molecular weight.

**Figure 7.2** Cellular convection for a liquid. For gases, due to their different temperature viscosity relationship, the gas falls in the centre of the cell.
FIGURE 7.3  OBSERVATION OF CELLULAR CONVECTION

FIGURE 7.4  BASE FLOW BETWEEN INCLINED PLATES
**FIGURE 7.5**
Local heat transfer coefficient between two inclined plates (see Figure 7.6). Source: Norman, C. and Roman, Y. 'Convecitive instability: A physicist's approach,' Journal of Modern Physics, Vol. 49, No. 3, July 1977.

**FIGURE 7.6**
Schematic depicting effect of gap spacing on conduction.
FIGURE 7.2  
PLLOT OF $h_c$ VERSUS PLATE SEPARATION $s$. $T_{wall} = 160^\circ C$, $T_{air} = 325^\circ C$.

FIGURE 7.8  
$h_c$ VERSUS TILT ANGLE TO THE HORIZONTAL FOR AIR ABSORPTION FOR VARIOUS ABSORBER TEMPERATURES ($T_a$) WITH COVER TEMP = 100°C.
Figure 7.9 Heat transfer coefficient variation with absorber temperature for convection and radiation.
Figure 7.10  True and predicted heat loss between two parallel plates 5 x 5 cm
Cover temperature 10 °C
**Figure 7.11**

Effective Rayleigh number versus molecular weight for different gases, at atmospheric pressure between two parallel plates, spacing 5 cm, 0.05 m, cold plate temperature 10°C, hot plate 30°C.
FIGURE 7.12  Heat transfer coefficient for gases of different molecular weight, for $S = 5 \text{ cm}$, cold plate temperature $10^\circ\text{C}$, hot plate temperature $30^\circ\text{C}$. 
Figure 7.13: Cost vs. Heat Transfer Coefficient for different gases. $f = 5\, \text{cm}$, volume of gas required for each square metre on collector is 50 litres.
**FIGURE 7.14** VARIATION OF HEAT TRANSFER COEFFICIENT $h_c$ WITH PRESSURE FOR A FLAT PLATE COLLECTOR, $S = 5\text{ cm}$, $T_i = 293\text{K}$, $T_2 = 823\text{K}$ FOR CURVE 1, $273\text{K}$ FOR CURVE 2 AND $473\text{K}$ FOR CURVE 3.

**FIGURE 7.15** DESCRIPTION OF TWO COVER SYSTEM.
FIGURE 7.16  VARIATION OF HEAT TRANSFER WITH GAP ACROSS A TWO COVER AND A SINGLE COVER SYSTEM. SOURCE: HONKOTA, A AND GARG, H. P. "MINIMIZING CONVECTIVE HEAT LOSSES." SOLAR ENERGY VOL. 25, NO. 6, P. 523.

FIGURE 7.17  REFLECTED SOLAR RAYS FOR A MULTI COVER SOLAR COLLECTOR.
**Figure 7.18** A solar ray and cut-away diagram of a hexagonal honeycomb collector. Source: Hollands, K.G.T. 'Advanced non-concentrating solar collectors.' Solar Energy Conversion Ed. by A.E Dixon and J.D. Leslie. Pergamon Press 1979.
FIGURE 7.19  HEAT TRANSFER COEFFICIENT $h_a$ DUE TO NATURAL CONVECTION FOR AIR AT ATMOSPHERIC PRESSURE BETWEEN TWO PARALLEL FLAT PLATES SPACING $5\text{cm}$, $T_i = 283\text{K}$, WITH A HONEYCOMB PAD WITH SLATS ASPECT RATIO 5
Figure 7.20 Thermal conductivity versus Rayleigh number for various gases $T_1 = 10^\circ C$, $T_2 = 80^\circ C$, $S = 5$ cm.
Figure 7.21 Rayleigh Number versus Temperature for Argon and Air at Atmospheric Pressure Between Two Parallel Flat Plates. Spacing $s = 5$ cm, Cold Plate Temperature $T_c = 10^\circ$C.
Figure 7.22. Heat transfer coefficients for several collector configurations.

- , Convection and conduction, air, atmospheric pressure, tilt angle 0°.
- , Convection and conduction, air, atmospheric pressure, tilt angle 60°.
- , Convection and conduction, argon, atmospheric pressure, tilt angle 0°.
- , Convection and conduction, air, honeycomb, tilt angle 60°.
- , Radiation, T max = 20°C, T min = 0°C, glass cover (ε = 0.08).
- , Conduction only, air at 4 × 10^3 Pa.
- , Conduction only, argon at 3 × 10^3 Pa.

S = 5 cm, T = 10°C.
FIGURE 7.23  GUARD RING HEATER

FIGURE 7.24  GUARD RING UNBALANCE VERSUS MEASURED HEAT TRANSFER ACROSS A 5cm THICK STYROFOAM SAMPLE
FIGURE 7.25 ACRYLIC TEST PANEL

FIGURE 7.26 SCHEMATIC DIAGRAM OF GUARDED HOT PLATE APPARATUS.
To Temperature Controlled Water Bath

15mm Diameter Cu Pipe

3mm Cu Sheet

Figure 7.27 Copper Cold Plates.
Figure 7.28 Measured and theoretical heat transfer coefficients for different gases between two parallel plates, $s = 5\,\text{cm}$ versus temperature difference.
FIGURE 7.29 THEORETICAL AND MEASURED HEAT TRANSFER $h_c$ FOR AIR AND ARGON
FIGURE 7.30  THEORETICAL HEAT TRANSFER ACROSS STRUCTURED POLYCARBONATE OF VARIOUS THICKNESSES, BOTH RADIATION AND CONVECTION, ASSUMING FLAT CONVECTION AND A MEASURED EMISIVITY OF 0.72.
PLATE 2.1
PROTO PROMETHEUS, 1. COLLECTOR, 2. STORE TOP INSULATION
AND COLLECTOR RRR INSULATION, 3. FAN MOTOR 4. MONITORING
EQUIPMENT, 5. SPACE FOR INSULATION.
PLATE 2.2  PROTO PROMETHEUS STORAGE TANK FILLED WITH PEBBLES.
PLATE 5.1

SOLAR SIMULATOR TESTING A STRUCTURED POLYCARBONATE COLLECTOR.

17. STRUCTURED POLYCARBONATE COLLECTOR, 16. WIND GENERATOR.

14. COOL RAY LAMPS.
PLATE 5.2

INDOOR COLLECTOR TEST FACILITY

7. DATA LOGGER, 8. STRUCTURED POLYCARBONATE COLLECTOR,
9. PRESSURE TAPS, 10. SITE OF ORIFICE PLATE
PLATE 7.1  VIEW OF HEATED OIL FILM FROM AN INFRARED CAMERA. THE BRIGHTER THE SPOT THE HOTTER THE SPOT.
PLATE 7.2  GUARDED HOT PLATE THERMAL CONDUCTIVITY RIG
11. INSULATED GUARD RING AND TEST CELL, 12. GAS CYLINDER
13. WATER COOLER, 14. HEATER POWER SUPPLY
APPENDIX A

SUNSTORE: Computer model of interseasonal store and sample output.
10 REM ******************** SUNSTORE ********************
20 REM
30 SHORT DEMAND(12)
40 SHORT SOL(12,24)
50 ASSIGN 1 TO "SUN DATA"
60 READ 1 TO SOL(12,24)
70 SHORT TEM(12,24)
80 ASSIGN 2 TO "TEM DATA"
90 READ 2 TO TEM(12,24)
100 DIM MONTHS(12,13)
110 ASSIGN 3 TO "MONTHS"
120 READ 3 TO MONTHS(12,13)
130 SHORT DAYS(12)
140 ASSIGN 4 TO "DAYS"
150 READ 4 TO DAYS(12)
160 PRINT USING 200
170 TOTSUN=W
180 PRINT "=TOTAL ANNUAL SOLAR RADIATION" ; W
190 PRINT """"""""""""""""""""""""""""""""""
200 IMAGE ///////////////////////////////////////////////////////////////////// SOLAR RADIATION AT KEN DISTRIBUTION OF HOUMLY GLOBAL IRRIGATION /////////////////////////////////////////////////////////////////////
210 PRINT USING 220
220 IMAGE "********** ON A HORIZONTAL SURFACE IN MJ/m2 ********************
230 FOR H=1 TO 12 ! print month heading
240 PRINT TAB (6*M):MONTHS(M)
250 NEXT M
260 FOR H=1 TO 24 ! print month heading
270 PRINT TAB (6*H):SOL(M,H)
280 PRINT TAB (M6):SOL(M,H)
290 TOTSUN=TOTSUN+SOL(M,H)*DAYS(M) ! calculate total annual solar radiation.
300 NEXT M
310 PRINT TAB 100 !
320 NEXT H
330 PRINT "TOTAL ANNUAL SOLAR RADIATION = "TOTSUN,"MJ/m2"
340 REM **************************** DATA INPUT*******************************
350 REM
360 REM
370 REM
380 REM
390 REM
400 REM
410 REM
420 REM
430 REM
440 REM
450 REM
460 REM
470 REM
480 REM
490 REM
500 REM
510 REM
520 REM
530 REM
540 REM
550 REM
560 REM
570 REM
580 REM
590 REM
600 REM
610 REM
620 REM
630 REM
640 REM
650 REM
660 REM
670 REM
680 REM
690 REM
700 REM
710 REM
720 REM
730 REM
740 REM
750 REM
760 REM
770 REM
780 REM
790 REM
800 REM
810 REM
820 REM
830 REM
840 REM
850 REM
860 REM
870 REM
880 REM
890 REM
900 REM
910 REM
920 REM
930 REM
940 REM
950 REM
960 REM
970 REM
980 REM
990 REM

1000 REM
1010 PRINT TAB (140);"MONTHS(1);":=":DEMAND(1);": print heating demand each mon
1020 TOTD=TOTD+DEMAND(1); calculate total annual heating dem.
1030 DEMAND(1)=DEMAND(1)/ICOLAREA/HOUSE; heating demand per m2 of collector
1050 PRINT "TOTAL EM... HOUSE PER ANNUM ";TOTD/1000;"GJ";";TOTDI
1060 PRINT "
1070 PRINT USING 1080 "
1080 IMAGE // SYSTEM OPERATION
1090 PRINT "ITH = Threshold Level (collector will only operate above this ini
tial) (W/m2)"
1100 PRINT "Tso = Original Store Temperature at the beginning of month (C)"
1110 PRINT "Ta = Ambient Temperature Averaged over periods of collector operat
1120 PRINT "Tt = Time Period of Collector Operation (Ms)"
1130 PRINT "It = Total Radiation which is above Threshold (MJ/m2)"
1140 PRINT "qN = Normalized Net Heat to Storage =qT-1m-s (MJ/m2)"
1150 PRINT "qT = Usefull Heat Collected= qN+1m (MJ/m2)"
1160 PRINT "qM = Normalized Total Monthly Load (MJ/m2)"
1170 PRINT "qN = Normalized Total Monthly Store Loads (MJ/m2)"
1180 PRINT "qAUX = qN+qTAUX"
1190 PRINT "THSOL=qN+qTAUX+qM+qAUX"
1200 PRINT "
1210 M ITH Tso Ta Tt It qN ts4 qt im qAUX "
1220 PRINT "
1230 FOR I=1 TO 12
1240 TSO=0
1250 TEMP=0
1260 J=1 TO 24
1270 IF ITH+TTS=0 THEN GOTO 1330
1280 IF ITH=0 THEN GOTO 1350
1290 IF ITH+TTS=0.056 THEN GOTO 1330
1300 TEMP=1000000
1310 TEMP=TEMP+TTS
1320 TEMP=TEMP/TTS
1330 NEXT J
1340 IF TEMP THEN GOTO 1360
1350 TEMP=TEMP/TTS
1360 TEMP=TEMP#3600/DAYS(I)
1370 TEMP=TEMP+TTS
1380 IF TEMP AND COUNT=0 THEN GOTO 1770
1390 TEMP=3600*24*2DAYS(I)/1000000
1400 TEMP=TEMP+TTS
1410 qT=FIIL(TSS-1TLUL-(TSS-TEMP)*E)
1420 I=USATs(TSS-TFt)*E
1430 qN=qT-1m-s=USATs(TSS-TFt)*E
1440 IF TFt<1500 THEN GOTO 1530
1450 IF TFt<1500 THEN GOTO 1530
1460 IF TFt<1500 THEN GOTO 1530
1470 IF TFt<1500 THEN GOTO 1530
1480 IF TFt<1500 THEN GOTO 1530
1490 IF TFt<1500 THEN GOTO 1530
1500 IF TFt<1500 THEN GOTO 1530
1510 NEXT I
1520 GOTO 1540
1530 Tsean=(TSS-TFs)/2
1540 I=USAs(Tsean-TTs)
1550 qN=qT-1m-s=USAs(Tsean-TTs)
1560 qN=qT-1m-s=USAs(Tsean-TTs)
1570 qN=qT-1m-s=USAs(Tsean-TTs)
1580 COUNT=1
1590 qAUX=qN+QTAUX
1600 EXTRA=0
1610 IF TFt>10 THEN qAUX=0
1620 PRINT USING 1960 ;MONTHS(1),ITH,TEMP,TT,TSOL,qAUX,qT,USATSE,HEM
1630 X=X+10
1640 Y=TSO/2
1650 PLOT X,Y
1660 TOTAL=TOTAL+TSOL
1670 qT=qT+QTAUX
1680 qT=qT+QTAUX
1690 qT=qT+QTAUX
1700 IF I>1 THEN I=10
1710 NEXT I
1720 QD=(QD+QD)*21 THEN GOTO 1780
1730 TSOL=TFS
1740 IF I=12 THEN I=0
1750 NEXT I
1760 TSSOL=QD/1000000
1770 NEXT I
1780 QD=QD+QD
1790 QD=QD+QD
1800 PRINT "THSOL=QD/1000000";
1810 PRINT USING 1810 ;QD+QD,TSOL,TEMP,TT,TSOL,qAUX,qT,USATSE,HEM
1820 PRINT "";
1830 SUN=(1-(QD+QD))/1000;"% energy supplied by solar system";
1840 PRINT USING 1840 ;% OF ENERGY SUPPLIED BY SOLAR SYSTEM";
1850 PRINT USING 1850 ;% OF SOLAR ENERGY COLLECTED ABOVE THRESHOLD";
1860 PRINT USING 1860 ;% OF SOLAR ENERGY COLLECTED";
1870 PRINT "TOTAL AUXILIARY FOR SYSTEM";
1880 PRINT "TOTAL AUXILIARY ENERGY FOR HOUSE";
1890 PRINT "";
1900 PRINT "";
1910 PRINT "";
1920 PRINT "";
1930 PRINT "";
1940 PRINT "";
1950 PRINT "";
1960 PRINT "";
1970 PRINT "";
1980 PRINT "";
1990 PRINT "";
2000 PRINT "";
2010 DATA 7750,6490,5560,3320,980,770,770,770,770,770,5270,7450
2020 END
### Solar Radiation at New Distribution of Hourly-Global Irradiation

**On a Horizontal Surface in MJ/m²**

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<th>FEB</th>
<th>MAR</th>
<th>APR</th>
<th>MAY</th>
<th>JUN</th>
<th>JUL</th>
<th>AUG</th>
<th>SEP</th>
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<td>.27</td>
<td>.28</td>
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**Total Annual Solar Radiation:** 3410.94 MJ/m²

---

**System Operation**

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<th>Tso</th>
<th>It</th>
<th>qN</th>
<th>tsf</th>
<th>qT</th>
<th>It</th>
<th>qN</th>
<th>tsf</th>
<th>qT</th>
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<td>198.57</td>
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**Total:** 1026.80 1488.93 390.2 -462.

% of Energy Supplied by Solar System: 69.0%
% of Solar Energy Collected Above Threshold: 42.0%
% of Solar Energy Collected: 41.5%

**Total Auxiliary Energy for System: 1293960.78469 MJ (-33543.3351803 KWh)

**Auxiliary Energy per House: 1293960.6078649 MJ (13594.3351803 KWh)**

---

**Collector**

**Store**

- **Store Length:** 280 Meters
- **Store Width:** 10 Meters
- **Store Height:** 4 Meter
- **Storage Material Pebbles Density:** 1600 kg/m³
- **Specific Heat:** 0.837 KJ/Kg°C
- **Store Insulation Thickness:** .6 m
- **Thermal Conductivity:** 0.056 W/m°C

**Collector**

- **Total Collector Area:** 2800 m²
- **Ft Heat Transfer Factor (equivalent to Fr heat removal factor if store has a good heat exchanger):** .9
- **UL Overall Heat Loss Coefficient:** 1
- **Ta Optimal Efficiency Averaged over Useful Incident Angles:** .8

---

**House**

- **Number of Houses:** 100
- **The Monthly Heating Load for Each House is (heating and hot water):** MJ
  - Jan: 7750
  - Feb: 6490
  - Mar: 5560
  - Apr: 3320
  - May: 980
  - Jun: 770
  - Jul: 770
  - Aug: 770
  - Sep: 770
  - Oct: 1790
  - Nov: 5270
  - Dec: 7450
- **Total Energy Demand of House per Annum:** 41.69 GJ (11580.555556 KWh)
Computer models used to predict steady state performance of air heating collectors.

**TOPAIR:** calculates the top heat loss $U_t$ for different absorber temperatures.

**EFFIC:** Calculates the efficiency of a top duct air heating collector.

**EFFIC2:** Calculates the efficiency of a rear duct air heating collector.
10 REM ************** PROGRAM TOPAIR ***************
20 REM ************** CALCULATE THE TOP LOSS COEFFICIENT FOR A SINGLE GLASS TIP/TAPE
30 REM
40 '50 FOR I=0 TO 20
60 TP=10+I85 'ABSORBER TEMP
70 TA=10 'Ambient temp (C)
80 WIND=1 'Wind speed (ms-1)
90 ERP=.95 'Absorber emissivity
100 EC.B='Cover plate emissivity
110 S=5 'Plate separation (cm)
120 G=9.81 'Acceleration due to gravity (ms-2) at LONDON
130 K=.0257 'Thermal conductivity of gas at Tave (W/m-20C)
140 B=0 'Tilt angle (Horizontal)
150 CPw=1007 'Heat capacity of air (J/KG)
155 CPw=1007 'Heat capacity of GAS BETWEEN COVER AND ABSORBER KG
160 S=S/100 'CONVERT TO METERS
170 L=1
180 W=1
190 SW=2.4LW/(L+W)
200 REM ************** THERMAL VOLUME EXPANSION COEFFICIENT ONLY HOLDS FOR PERFE
210 TC=TA+(TP-TA)/2 'guess the cover temp
220 TP=273.15+TC 'CONVERT TO KELVIN
230 TA=273.15+TC 'CONVERT TO KELVIN
240 TC=273.15+TC 'CONVERT TO KELVIN
250 TP=TP+273.15 'CONVERT TO KELVIN
260 Tc=TP 'CONVERT TO KELVIN
270 DT=TP+1 'TEMP DIFFERENCE DELTA T
280 Tave=(TP+Tc)/2 'AVERAGE GAS TEMPERATURE
290 DEN=352.91/Tave
300 k=Tave.000074+.0034406
310 VIS=Tave..0000046+.0000046351
320 VOL/Tave='THERMAL VOLUME EXPANSION COEFFICIENT ONLY HOLDS FOR PERFE
330 VWVIS/DEN='KINEMATIC VISCOSITY
340 Gr=GVOL*SW/360 'GRASHOF NUMBER
350 Pr=CPVIS/K
360 Rn=GrPr 'RAYLEIGH No
370 REM 'CALCULATE NUSLUB NUMBER
380 N2=(RatCOS (B)/5830)^(1/3-1)
390 IF N2 THEN N1=0 'TAKE ONLY POSITIVE TERMS
400 N2=(RatCOS (B)/5830)^(1/3-1)
410 IF N2 THEN N2=0 'TAKE ONLY POSITIVE TERMS
420 N1=1.4441(1-1/1/1.61708/RatCOS (B))+N2 'NUSLUB No
430 hc=k/SINU 'HEAT TRANSFER COEFFICIENT
440 hr=0.0000000679*(TP+2*TC*1/1/EPE+1/EC-1) 'RADD FROM PLATE TO COVER
450 hsky=0.0000000679*EC*(TC+2*TA*1/1/EPE+1/EC-1) 'RADD COVER TO SKY
470 DTw=TC-TP
480 Tave=TA_DTW/2
490 DENw=352.91/Tave
500 KH=Tave.000076+.0034406
510 VISw=Tave..0000046+.0000046351
520 REM 'TOP ASSIGNMENT
530 VWVISw/DENw='KINEMATIC VISCOSITY
540 Grw=GVOLw*Sww/360 'GRASHOF NUMBER
550 Prw=CPVISw/K
560 Rnw=GrwPr 'RAYLEIGH No
570 hnw=1.4441Nw*(TP+2*TC*1/1/EPE+1/EC-1) 'RADD FROM PLATE TO COVER
580 hnw=1.4441Nw*(TP+2*TC*1/1/EPE+1/EC-1) 'RADD COVER TO SKY
590 Tc=TP-UTP[TP-TC]*TP-TC 'CALCULATE COVER TEMP (C)
600 IF ABS (TC-TC)<0.01 THEN TC=TC ELSE GOTO 630
610 Tc=TC
620 GOTO 270
630 PRINT " TOP LOSS COEFFICIENT CALCULATION (see Duffie & Beckman pp204)"
640 PRINT " "
650 PRINT " "
660 PRINT " "
670 PRINT " "
680 PRINT " "
690 PRINT " "
700 PRINT " "
710 PRINT " "
720 PRINT " "
730 PRINT " "
740 PRINT " "
750 PRINT " "
760 PRINT " "
770 PRINT " "
780 PRINT " "
790 PRINT " "
800 PRINT " "
810 PRINT " "
820 PRINT " "
830 PRINT " "
840 PRINT " "
850 PRINT " "
860 PRINT " "
870 PRINT " "
880 PRINT " "
890 PRINT " "
900 PRINT " "
910 PRINT " "
920 END
"10 REM .......................... EFFICIENCY .........................."
20 REM THIS PROGRAM CALCULATES THE STEADY STATE EFFICIENCY OF A TOP DUCT
30 REM AIR HEATING SOLAR COLLECTOR USING EQUATIONS FROM DUFFIE AND BECKMAN
40 REM p237 Figure 6.12.1 (d)
50 REM
60 REM INPUT VARIABLE DATA
65 FOR J=0 TO 10
70 IF M<11.4+J*10 THEN M=1.4+J*10 ' MASS FLOW RATE (kg/hr)
80 TA=14.2 ' AMBIENT TEMP (°C)
90 TIN=T A ' IN FLUID TEMPERATURE (°C)
100 T2=20.4 ' ABSORBER TEMPERATURE (°C) IF THIS CHANGES ALSO CHANGE T1
110 T1=(T2-TA)/2+TA ' TEMPERATURES
120 WIND=5 ' WIND SPEED (m/s)
130 I=236 ' INTENSITY OF SOLAR RAD (W/m2)
140 E=8 ' TRANSMISSIVITY # ABSORBIVITY OF COVER AND ABSORBER
150 E2=.95 ' EMISSIVITY OF ABSORBER
160 H=0.034 ' CONDUCTIVITY OF REAR INSULATION (W/m°C)
170 Tm=0.075 ' INSULATION THICKNESS (m)
190 A1=1 ' COLLECTOR AREA (m2)
200 L=2 ' COLLECTOR LENGTH IN METERS
210 W=1 ' WIDTH OF COLLECTOR IN METERS
220 S=1 ' PLATE SEPARATION IN CM
230 D=1 ' FIN SEPARATION IN CM
240 DISP "DO YOU WANT ALL THE COLLECTOR PARAMETERS PRINTED Y OR N ????
245 IF Y=NO THEN GOTO 470
250 INPUT A$ ' A$="N" THEN GOTO 470
260 IF A$="N" THEN GOTO 470
270 PRINTER IS 701
280 PRINT " " COLLECTOR INITIAL PARAMETERS ARE
290 PRINT USING 930 ' "MASS FLOW RATE",M,"Kg/hr"
300 PRINT USING 930 ' "AMBIENT TEMP",TA,"°C"
310 PRINT USING 930 ' "INLET FLUID TEMP",TIN,"°C"
320 PRINT USING 930 ' "ABSORBER TEMP",T2,"°C"
330 PRINT USING 930 ' "WIND SPEED",WIND,"m/s"
340 PRINT USING 930 ' "SOLAR RADIATION",I,"W/m2"
350 PRINT USING 930 ' "TRANSMISSIVITY & ABSORBIVITY",Ta
360 PRINT USING 930 ' "EMISSIVITY OF COVER",E1
370 PRINT USING 930 ' "EMISSIVITY OF ABSORBER ",E2
380 PRINT USING 930 ' "INSULATION CONDUCTIVITY",K,"W/m°C"
390 PRINT USING 930 ' "INSULATION THICKNESS",Tm,"m"
400 PRINT USING 930 ' "COLLECTOR AREA",A1,"m2"
410 PRINT USING 930 ' "COLLECTOR LENGTH",L,"m"
420 PRINT USING 930 ' "COLLECTOR WIDTH ",W,"m"
430 PRINT USING 930 ' "PLATE SEPARATION",S,"cm"
440 PRINT USING 930 ' "FIN SEPARATION",D,"cm"
450 PRINT 
460 REM INPUT CONSTANT DATA
470 STE=.00000000567 ' STEFAN-BOLTZMANN CONSTANT (W/m2K4)
490 VIS=.0000186 ' VISCOSITY OF AIR IN DUCT (N-s/m2)
500 K=.0241 ' THERMAL CONDUCTIVITY OF AIR IN DUCT (W/mK)
510 C=1099 ' HEAT CAPACITY OF AIR AT CONSTANT PRESSURE (J/kg°C)
520 REM 
530 T1=11.273.15 ' ELVING
APPENDIX C

A multi-node dynamic computer model of a flat-plate, rear-duct, air-heating solar collector is described, and its verification is outlined. Results from the model are then presented of the daily averaged thermal efficiencies for a variety of simulated ambient conditions pertinent to mid to high maritime latitudes. The collectors differ significantly only in their thermal capacitances. The diurnal variation of insolation produces a modest spread of thermal efficiencies, the lower the thermal capacitance of the collector the higher the efficiency. More rapid fluctuations in insolation produce only a slightly further spread in the thermal efficiencies, though such fluctuations have a more significant effect on peak temperatures.

Keywords: air-heating solar collectors; thermal capacitance effects in solar collectors.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>DY1-5</td>
<td>plate and duct-back thicknesses (5)</td>
</tr>
<tr>
<td>( f(\theta) )</td>
<td>transmittance - absorbance function of the collector</td>
</tr>
<tr>
<td>( F_R )</td>
<td>collector heat-removal factor</td>
</tr>
<tr>
<td>HPA(I)</td>
<td>heat-transfer coefficient plate (or duct-back) to air in the I'th segment of the duct</td>
</tr>
<tr>
<td>M</td>
<td>duct air flow rate</td>
</tr>
<tr>
<td>NI</td>
<td>number of duct segments</td>
</tr>
<tr>
<td>PON</td>
<td>threshold power for switch on of air flow</td>
</tr>
<tr>
<td>S</td>
<td>irradiance in cover plane</td>
</tr>
<tr>
<td>S0</td>
<td>solar beam irradiance</td>
</tr>
<tr>
<td>S1</td>
<td>diffuse irradiance on a horizontal surface</td>
</tr>
<tr>
<td>SP</td>
<td>irradiance absorbed by plate</td>
</tr>
<tr>
<td>TA</td>
<td>ambient temperature</td>
</tr>
</tbody>
</table>
Low mass in solar collectors offers the advantage of low construction and installation costs. But the mass also influences the thermal capacitance and hence the thermal efficiency, because even a smooth diurnal variation of insolation prevents a collector from achieving a true steady-state, and the lower the mass the closer the varying conditions are followed. Earlier studies (for example {1}, {2}, {4}) have shown that lowering the mass will improve the thermal efficiency, though perhaps by not very much. However, there seem to be few data on the diurnal performance in various ambient conditions of collectors which differ only in their thermal capacitances. This is particularly the case for air-collectors.

Therefore we have developed and verified a dynamic computer model of a flat-plate, rear-duct, air-heating solar collector. We have used it to obtain daily averaged thermal efficiencies for a wide variety of simulated ambient conditions pertinent to maritime mid to high latitudes. The basic configuration of the collector was varied to yield a wide spread of thermal capacitances. The model is of the multi-node kind, because various studies (for example {1}, {3}, {4}) have shown that simple one-node models are unlikely to give accurate results in non steady-state conditions.

2 THE COLLECTOR MODEL

The collector is of the flat-plate rear-duct air-heating single-cover kind, with dimensions selected to give good performance. It is divided into nodes as shown in Figure 1. (This collector could be complete, or it could be a strip width W of a larger assembly.) Heat balance equations are defined at each node, and the equations are numerically integrated in sequence using the Adams-Bashforth-Moulton predictor-corrector method {5}.

The model was tested in a variety of ways, including a comparison of its predictions with the actual behaviour in the laboratory of a flat-plate rear-duct air-heating single-cover collector. In all cases the agreement between prediction and actuality was satisfactory.
RESULTS

5.1 The collectors

Table 1 specifies the collector configurations, and the rear-duct air flow conditions. The basic configuration was selected to give good steady-state performance, the configurations differing only in the thickness of the plate and duct-back (DY1 to DY5 in Table 1). The main effect of these changes in configuration is on the thermal capacitance of the components and hence of the whole collector.

Table 1 Collector configurations, and rear-duct air flow

<table>
<thead>
<tr>
<th>collector length (along flow)</th>
<th>4.00 m</th>
</tr>
</thead>
<tbody>
<tr>
<td>collector width (W)</td>
<td>1.00 m</td>
</tr>
<tr>
<td>cover to plate spacing</td>
<td>0.05 m</td>
</tr>
<tr>
<td>rear duct gap</td>
<td>0.01 m</td>
</tr>
<tr>
<td>back insulation</td>
<td>dry glass fibre, thickness 0.10 m</td>
</tr>
<tr>
<td>edge insulation</td>
<td>dry glass fibre, thickness 0.05 m</td>
</tr>
<tr>
<td>material of plate and duct-back</td>
<td>duralumin HS15TB</td>
</tr>
<tr>
<td>cover</td>
<td>polycarbonate, thickness 2.00 mm</td>
</tr>
<tr>
<td>plate absorbtance</td>
<td>0.95 at θ=0, falling slightly as θ increases</td>
</tr>
<tr>
<td>emissivity of upper surface of the plate (diffuse)</td>
<td>0.10</td>
</tr>
<tr>
<td>emissivity of duct surfaces (diffuse)</td>
<td>0.91</td>
</tr>
<tr>
<td>emissivity of the cover (diffuse)</td>
<td>0.85</td>
</tr>
<tr>
<td>thermal properties of air at 283 K for ambient air, at 303 K elsewhere</td>
<td></td>
</tr>
<tr>
<td>latitude</td>
<td>52°N</td>
</tr>
<tr>
<td>collector tilt (to horizontal)</td>
<td>35°</td>
</tr>
<tr>
<td>collector orientation</td>
<td>south-facing</td>
</tr>
<tr>
<td>thickness of plate and of duct-back</td>
<td>collector time-constant (flow 1)</td>
</tr>
<tr>
<td>DY1</td>
<td>0.2 mm</td>
</tr>
<tr>
<td>DY2</td>
<td>0.5 mm</td>
</tr>
<tr>
<td>DY3</td>
<td>1.0 mm</td>
</tr>
<tr>
<td>DY4</td>
<td>2.0 mm</td>
</tr>
<tr>
<td>DY5</td>
<td>5.0 mm</td>
</tr>
</tbody>
</table>

Air flow in the rear-duct

<table>
<thead>
<tr>
<th>flow 0</th>
<th>stagnation (M=0)</th>
</tr>
</thead>
<tbody>
<tr>
<td>flow 1</td>
<td>all TI M = 0.0600 kg s⁻¹ (PON, irrelevant)</td>
</tr>
<tr>
<td>flow 2</td>
<td>TI = 303 K M = 0.0600 kg s⁻¹ PON = 128 W</td>
</tr>
<tr>
<td>flow 3</td>
<td>TI = 323 K M = 0.0562 kg s⁻¹ PON = 124 W</td>
</tr>
</tbody>
</table>

The air flow rate is a compromise between attaining large values of HPA(I) and keeping low the power required to maintain the air flow in the rear-duct. At \( M = 0.0600 \) kg s⁻¹ and \( TI = 303 \) K (flow 2 in Table 1) this power is 6.4 W. The corresponding pressure drop across the duct is 12 mm water gauge. If it is
assumed that the circulation fan gives a constant volumetric flow rate then at
other values of $T_1$ the value of $M$ will be different from $0.0600 \text{ kg s}^{-1}$: at
$T_1 = 323 \text{ K}, M = 0.0562 \text{ kg s}^{-1}$ (flow 3 in Table 1).

It is also necessary to specify the minimum power that must be delivered by a
complete array of collectors in order for the air flow to either be switched
on or be sustained. This power must be some multiple of the electrical power
required by the fan to circulate air around the whole system incorporating the
array. We adopted a multiple of two. In order to estimate the electrical
power it is necessary to allow for the efficiency of the fan and for the pressure
drop in the whole system. For a modest domestic system we ended up with a
minimum power per collector of the sort specified in Table 1 of 128 W for flow 2.
For flow 3 $P_{ON}$ is slightly less. The values of $P_{ON}$ are shown in Table 1. Note that
the values of $P_{ON}$ are for a 4 m x 1 m collector, and not for the whole array.
These values of $P_{ON}$ correspond to an air temperature rise of between 2 K and 3 K
for the flow conditions specified.

The collector time-constants in Table 1 vary with ambient conditions and with
operating conditions, particularly with the air flow rate. The values in the
Table are representative for all ambient conditions considered here, and for
the various (similar) air flow rates, except for flow 0 (stagnation), in which
case the time-constants in Table 1 should be multiplied by about a factor of 5.
Note that the time-constants in Table 1 are the $1/e$ time-intervals following a
step change in insolation. However, only in stagnation is the response very close
to exponential. Note also that the thermal capacitance of the cover has a
relatively small effect, because the cover is coupled to the plate via a rather
large thermal resistance.

3.2 Steady-state efficiency curve

We obtained a standard steady-state thermal efficiency curve, of the form (6)
\[
\eta = F_R \left( f(\theta) - U_L (T_1 - T_A)/S \right) 
\]
where $f(\theta)$ is such that
\[
S \eta = f(\theta) \cdot S 
\]
For the steady state efficiency curve $S$ is beam irradiance normal to the cover,
such that $S = 700 \text{ W m}^{-2}$. Furthermore, $T_A = 293 \text{ K}, T_K = 273 \text{ K}, WIND = 1.0 \text{ m s}^{-1}$,
$M = 0.0600 \text{ kg s}^{-1}$. These values lie within the ASHRAE specifications for steady-
state collector testing (6).

In order to obtain the efficiency curve the value of $T_1$ was varied, everything
else remaining constant. The outcome is shown in Figure 2 for collector
configuration DY1 (Table 1), though the results for DY2 to DY5 are
indistinguishable from those for DY1 on the scale of Figure 2. The intercept
on the $\eta$-axis, 0.683 gives $F_R \cdot f(\theta)$ (equation (1)). The program yields a value of
0.830 for $f(\theta)$, and therefore $F_R$ is 0.823. The slope gives $-F_R \cdot U_L$, and at low
values of $(T_1-T_A)/S$ this is $-2.83 \text{ W m}^{-2} \text{ K}^{-1}$, giving a value of $U_L$ of 3.44 $\text{ W m}^{-2} \text{ K}^{-1}$.
The value of $F_R \cdot U_L$ increases as $T_1$ increases ($T_A, S$ constant), largely because
the radiative heat transfer coefficients increase with increasing temperature
differences, and though $F_R$ decreases it does not offset the increase in $U_L$.
These values of $f(\theta)$, $F_R$ and $U_L$ indicate good performance for a flat-plate rear-
duct air-heating single-cover collector with a selective plate-surface.

We had a "quick look" at the effect of varying the wind speed on the steady-state
temperatures. The effect was fairly modest, because of the large thermal resistance between cover and plate. Wind speed variations will be deferred to a later study.

3.3 Daily-averaged efficiency

The collector configurations DY1 to DY5 were run under conditions flow 2 and flow 3 for a variety of simulated days 21 June (J), 21 March (M), 21 December (D). The simulated conditions of insolation and weather on these days are shown in Figure 3. The ambient temperature $T_A$ varies sinusoidally through the day (Figure 3(a)) with an amplitude of 5 K. Note that there are two temperature curves for 21 December, $T_{A1}$ and $T_{A2}$. The irradiance $S$ consists of a diffuse component from the ground, and of a sky component which can either correspond to clear sky conditions or to overcast diffuse conditions. Figure 3(b) shows some of the various insulations, the prefix $S_0$ denoting the clear sky irradiance normal to the beam, and the prefix $S_1$ the overcast diffuse irradiance on a horizontal surface. In the cases in Figure 3(b) the only variation in insolation is the diurnal envelope shown. By contrast in Figures 3(c) and (d) the insolation flips between the two envelopes shown, the square wave periods being indicated, the conditions remaining diffuse throughout. In clear sky conditions the sky temperature is 20 K below $T_A$, and in overcast conditions it is 10 K below $T_A$. In all cases the wind speed is constant at 1.0 m s$^{-1}$.

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

$$\bar{\eta} = \frac{\text{total energy extracted by the air flow in the day/integration of } S\text{ over the day}}{S}.\quad (3)$$

Note that a day spans the time from sunrise to sunset. In no case did a collector deliver energy before or after sunset, and therefore $\bar{\eta}$ is never being wrongly evaluated.

In order to plot $\bar{\eta}$ on Figure 2 it is necessary to re-define the abscissa $(T_I - T_A)/S$. $T_I$ is constant (303 K or 323 K), and for $T_A$ and $S$ the arithmetic mean values for the period sunrise to sunset are taken. The outcome is shown in Figure 2, the results being coded in accord with Table 1 and Figure 3, except that the thermal capacitance configuration DY1 to DY5 is not shown. However, you can see that at each value of $(T_I - T_A)/S$ there is a column of results, and in every case DY1 is at the top, then comes DY2, and so on, to DY5, though in some cases DY1-DY3 merge on the scale of Figure 2. Clearly, the lower the thermal capacitance the better the performance.

Consider first those cases in which the insolation only varies over the diurnal envelope: this covers the cases (i)-(vi), (viii), (xi). The increase in $\bar{\eta}$ is marked in going from the rather massive DY5 to the rather less massive DY4. However, the improvement in going from DY4 to the low mass DY1 is also significant, particularly in marginal conditions (large $(T_I - T_A)/S$). This general improvement with reducing thermal capacitance 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. Note that the sinusoidal variations in $T_A$ and $T_K$ do not make an appreciable contribution to the spread of $\bar{\eta}$ with thermal capacitance on the scale of Figure 2.

The advantage of low mass could, in principle, be more marked under intermittent irradiance. S1D1-S1D3 provide such conditions (Figure 3), the periodicities lying within the range of time-constants in Table 1. However, Figure 2 shows that, even in marginal conditions, very little further advantage in low mass is obtained, though DY1-DY3 are more spread out than with the diurnal envelope alone. The
reason for such a slight improvement is that whereas a low mass collector will "follow" the insolation, possibly switching the air flow on and off, a high mass collector, once it has warmed to the point where the air flow switches on, will tend to stay at a fairly constant temperature. The overall effect, for a wide variety of conditions, is that the time-averaged temperatures 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 \cite{1}.

Figure 2 also shows that the values of $\tilde{\eta}$ differ from those of $\eta$. This is particularly the case at low thermal capacitances, as can be seen from the performance of DY1, which is not very different from that which would have been obtained for a collector of zero thermal capacitance. Two prominent and opposing effects operating here are that for $\eta$ in Figure 2 the value of $\dot{E}$ is always zero, thus raising $f(\dot{E})$, and, more importantly, that in insolation conditions which vary, intermittently or otherwise, a collector can "grab" peak insolation, yet entirely miss the corresponding steady state insolation which never reaches such peak values. Low thermal capacitance is again an advantage.

In addition to $\tilde{\eta}$, the daily average of $T_0$ was also obtained, such that only those periods were included in which air flowed in the rear duct. In general the lower the thermal capacitance of the collector the higher the daily average, though the improvement from DY5 to DY1 never exceeded 2 K. However, the peak temperatures for DY1 can be up to about 10 K higher than for DY5, the greatest difference occurring in intermittent conditions. In some circumstances this will be an important advantage of low thermal capacitance.

A set of results analogous to those in Figure 2 was obtained for lower flow rates, around 0.02 kg s$^{-1}$. This is a potentially useful domain, because in spite of the lower thermal efficiencies the values of $T_0$ are raised and can reach values such that useful energy can be extracted from ambient conditions which would yield no useful energy at higher flow rates, because of the lower values of $T_0$. However the variation of $\tilde{\eta}$ with thermal capacitance (DY1-DY5) was not remarkably different from that shown in Figure 2.

It can be concluded that collectors with low thermal capacitance can have significantly larger thermal efficiencies at non-small daily averaged values of $(T_1-T_A)/S$ in non-steady insolation, and that this is largely because of the diurnal variation, rather than because of more rapid fluctuations in insolation. Peak temperatures can also be significantly larger at low thermal capacitance, particularly when there are rapid fluctuations in insolation.

REFERENCES

2. M. Yusoff and D. J. Close, Transient studies of solar air heaters, presented at the Inter-regional symposium on solar energy for development, Tokyo 5-10 February (1979).
Figure 1  Flat-plate, rear duct, air heating solar collector.

- nodes in airflow
- other nodes
Figure 2 Steady-state efficiency ($\eta$ - the solid curve) and daily averaged efficiency ($\bar{\eta}$). The values of $\bar{\eta}$ are for a variety of simulated conditions (see Table 1 and Figure 3).

(i) S0J/TAJ, flow 2  (ii) S0M/TAM, flow 2  (iii) S0D/TAD1, flow 2
(iv) S0M/TAM, flow 3  (v) S1M/TAM, flow 2  (vi) S0D/TAD1, flow 3
(vii) S1D1/TAD1, flow 2  (viii) S0D/TAD2, flow 3  (ix) S1D2/TAD1, flow 2
(x) S1D3/TAD1, flow 2  (xi) S1D/TAD1, flow 2.
Figure 3 Simulated ambient conditions. For further details see text.
APPENDIX D

TRANS: Computer programme for analysing collector data under transient conditions.
**FUNCTION**

Calculate the Collector Eff. from Trajectory Data

**DIMENSION**

N = 10

**REMARKS**

0. For a 50 ltr. collector

1. For ZT=1 to 10

2. For ZS=1 to 10

3. IF 100 = 0 THEN GOTO 650

4. IF 100 = 1 THEN GOTO 110

5. IF I = 0 AND X(NK) = 0 THEN GOTO 650

6. IF I = 1 THEN GOTO 110

7. IF I = 1 THEN GOTO 110

8. IF I = 1 THEN GOTO 110

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87. IF I = 1 THEN GOTO 110

88. IF I = 1 THEN GOTO 110

89. IF I = 1 THEN GOTO 110

90. IF I = 1 THEN GOTO 110

91. IF I = 1 THEN GOTO 110

92. IF I = 1 THEN GOTO 110
214
940 NEXT K
950 NEXT K
960 Z.old = S.0r (ZE.YY/(N.F-NC))
970 PRINT "ETAED=:E,.*-*:ZE"
980 U= X.(NC)
990 PRINT "F";U."":*/":Z.(NC)
1000 PRINT "TABLE 4.4"
1010 FOR K+1 TO N
1020 C(K)*X((H)/E
1030 PRINT K,C(K)
1035 NEXT K
1040 F=(U/(mLOG(I-U/H)))
1050 PRINT "F";I
1060 E=E/F
1070 U=U+F
1080 PRINT "ETAED=:E,";U":U"
1090 PRINT "DATA SETS ACCEPTED FOR ANALYSIS":N.P
1100 REM READ DATA TO GENERATE THERMAL PERFORMANCE CURVE
1110 ASSIGN 1 TO "TRANSD700"
1120 N.P=O
1130 READ# 1 : I,X(NK),Y,T(NK)
1140 IF I=O AND X(NK)=O THEN GOTO 1570
1150 I=I+1
1160 FOR K+2 TO NK
1170 L=NK+K+1
1180 READ# 1 ; I,X(L),Y,T(L)
1190 IF I=O AND X(L)=O THEN GOTO 1570
1200 IF I=1 THEN GOTO 1130
1210 I=I+1
1220 NEXT K
1230 GOTO 1400
1240 FOR K+2 TO NK
1250 L=NK+K+2
1260 X(L)=X(L)-1
1270 T(L)=T(L)-1
1280 NEXT K
1290 READ# 1 ; I,X(I),Y,T(I)
1300 IF I=O AND X(I)=O THEN GOTO 1570
1310 IF I=1 THEN GOTO 1150
1320 I=I+1
1330 E=E
1340 X=NC=O
1350 FOR K=1 TO NK
1370 Y=Y+F(E)
1380 PRINT Y,V(NC)
1390 REM CALC LEAST SOR TO THERMAL PERFORMANCE
1400 S.X=S.X+X(NC)
1410 S.Y=S.Y+Y
1420 S.Y=S.Y+Y
1430 S.Y=S.Y+Y
1440 S.Y=S.Y+Y
1450 S.Y=S.Y+Y
1460 S.Y=S.Y+Y
1470 S.Y=S.Y+Y
1480 S.Y=S.Y+Y
1490 S.Y=S.Y+Y
1500 S.X=S.X+X(NC)
1510 S.Y=S.Y+Y
1520 S.Y=S.Y+Y
1530 S.Y=S.Y+Y
1540 S.Y=S.Y+Y
1550 S.Y=S.Y+Y
1560 GOTO 1220
1570 DNP=N.P
1580 PRINT "POINTS ON THERMAL PERFORMANCE CHARACTERISTIC":N.P
1590 PRINT "FROM LEAST SQUARES FITS EACH WAY"
1600 E=((SY*SY-SX*SY)/(DNP*SY-SX*SY))
1610 U=(SX*SY-DNP*SY)/(DNP*SY-SX*SY)
1620 PRINT "MINIMUM ETAED=:E,";U":U"
1630 E=(SY*SY-SX*SY)/(DNP*SY-SX*SY)
1640 U=(SX*SY-DNP*SY)/(DNP*SY-SX*SY)
1650 PRINT "MAXIMUM ETAED=:E,";U":U"
1655 NEXT NK
1660 STOP
1670 END