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

Thesis

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

Thesis submitted for the degree of Doctor of Philosophy in Energy Research at the Open University, September 1984

Volume 2

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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 \((J \text{ Kg}^{-1} \text{ °C}^{-1})\)

\(c_p\) Volume heat capacity at constant pressure \((J \text{ Kg}^{-1} \text{ °C}^{-1})\)

\(C_h\) 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 \((J \text{ m}^{-2})\)

\(f\) 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 \((W \text{ m}^{-2})\)

\(K_h\) 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 \((J)\)

\(M_C\) Storage heat capacity \((J \text{ °C}^{-1})\)

\(N\) Lifetime of hardware \((\text{years})\)

\(n\) Number of years

\(P_{VCh}\) 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 \((\text{°C})\)

\(T_{at}\) Ambient temperature averaged over periods when the radiation level is above the threshold \((\text{°C})\)

\(T_g\) Monthly average ground temperature \((\text{°C})\)

\(T_s\) Store temperature \((\text{°C})\)

\(\bar{T}_s\) Monthly average store temperature \((\text{°C})\)

\(T_{so}\) Store temperature at the beginning of the month \((\text{°C})\)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ΔT</td>
<td>Temperature change (°C)</td>
</tr>
<tr>
<td>tₘ</td>
<td>Total number of seconds in a month</td>
</tr>
<tr>
<td>tₜ</td>
<td>Total number of seconds collector is in operation in month, i.e. when radiation level is above threshold</td>
</tr>
<tr>
<td>Uₖ</td>
<td>Collector overall loss coefficient (W m⁻² °C⁻¹)</td>
</tr>
<tr>
<td>Uₛ</td>
<td>Storage tank heat loss coefficient (W m⁻² °C⁻¹)</td>
</tr>
<tr>
<td>V</td>
<td>Volume (m³)</td>
</tr>
<tr>
<td>ρ</td>
<td>Density (kg m⁻³)</td>
</tr>
<tr>
<td>(τα)</td>
<td>Monthly average transmittance-absorptance product</td>
</tr>
</tbody>
</table>
Nomenclature

Chapter 3

A_c  Collector area (m^2)
FR  Collector heat-exchanger efficiency factor
f  Fraction of monthly total demand met by solar energy
H_T  Monthly average daily radiation incident on the collector surface per unit area (Jm^-2)
L  Monthly total heating demand for space heating and hot water (J)
N  Days in month
T_a  Monthly average ambient temperature (°C)
T_{ref}  An empirically derived reference temperature (100° C)
t_m  Total number of seconds in a month
U_L  Collector overall loss coefficient (Wm^-2 °C^-1)
(\tau_a)  Monthly average transmittance-absorptance product
Nomenclature

Chapter 4

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

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

Dh  Characteristic length (m)

F'  Absorber plate (or collector) efficiency factor

FR  Collector heat removal factor

g  Acceleration of gravity (m/s²)

h₁  Convective heat transfer coefficient, duct top to heat transfer fluid (W/m²·°C)

h₂  Convective heat transfer coefficient, duct base to heat transfer fluid (W/m²·°C)

hr  Radiative heat transfer coefficient (W/m²·°C)

hw  Wind heat transfer coefficient (W/m²·°C)

H  Duct height (m)

I  Equivalent normal solar irradiance (W/m²)

k  Thermal conductivity (W/m·°C)

L  Collector length (m)

m  Mass flow rate of transfer fluid (Kg/s)

Nu  Nusselt number

Pr  Prandtl number

Qu  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/m²·°C)

Ue  Edge loss heat transfer coefficient (W/m²·°C)

UL  Collector overall heat transfer (loss) coefficient (W/m²·°C)
$U_t$ Top loss heat transfer coefficient (Wm$^{-2}$ °C$^{-1}$)

$V$ Wind velocity (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 (K$^{-1}$)

$\varepsilon_c$ Cover emissivity

$\varepsilon_p$ Absorber plate emissivity

$\eta$ Efficiency

$\mu$ Absolute (dynamic) coefficient of viscosity (Kg m$^{-1}$ s$^{-1}$)

$\rho$ Density (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 (m\(^2\))

\( A_C \) Collector area (m\(^2\))

\( c_p \) Volume heat capacity at constant pressure (J/kg°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 \( \tau_a \) 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\(^{-1}\))

\( m_t \) Mass flow rate of leak (Kg s\(^{-1}\))

\( M \) Fluid capacity of collector (Kg)

\( (mc)_e \) Effective heat capacity of collector (J °C\(^{-1}\))

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

\( Qu \) Energy per unit time, useful (W)

\( (Qu)_t \) 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)
\( T_{\text{in}} \) Measured fluid inlet temperature (°C)
\( T_m \) Mean fluid temperature \((T_e + T_i)/2\) (°C)
\( T_p \) Absorber plate temperature (°C)
\( T_{\text{ap}} \) Mean absorber temperature (°C)
\( T_{\text{sky}} \) Equivalent black body sky temperature (°C)
\( T^* \) Reduced temperature \((T_i - T_a)/I\) (\(\text{m}^2 \text{ °C w}^{-1}\))
\( U_L \) Collector overall heat transfer (loss) coefficient \((\text{Wm}^{-2} \text{ °C}^{-1})\)
\( V \) Wind velocity (\(\text{ms}^{-1}\))
\( \eta \) Efficiency
\( \tau_\alpha \) Product of the absorptance of the collector plate and the transmittance of the cover for normal irradiance.
\( \tau_C \) Collector time constant under flow conditions (s)
\( \tau_d \) Cut off time (s)
\( (\tau_\alpha)_e \) Effective transmittance absorptance product
\( (\tau_\alpha)_o \) Product of the absorptance and transmittance for normal irradiance
\( \Delta T^* \) Time increment
\( \theta \) 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 ($W m^{-2} °C^{-1}$)
$h_{rp-c}$ Radiation coefficient between absorber plate and cover ($W m^{-2} °C^{-1}$)
$h_{rc-a}$ Radiation coefficient from the cover to sky ($W m^{-2} °C^{-1}$)
$h_w$ Wind heat transfer coefficient. ($W m^{-2} °C^{-1}$)
$I$ Equivalent normal solar irradiance ($W m^{-2}$)
$I_{th}$ Threshold solar irradiance ($W m^{-2}$)
$T_a$ Ambient air temperature ($°C$)
$T_i$ Inlet fluid temperature ($°C$)
$U$ Collector heat loss coefficient $F'U_L$ ($W m^{-2} °C^{-1}$)
$U_L$ Collector overall heat transfer (loss) coefficient ($W m^{-2} °C^{-1}$)
$\varepsilon_t$ Thermal emissivity
$\eta$ Efficiency steady state
$\bar{\eta}$ Daily averaged efficiency
$\eta_0$ Zero loss collector efficiency, $F'(\alpha \tau)$.
$\tau_s$ Solar transmissivity
$(\tau \alpha)$. Product of the absorptance and transmittance for normal irradiance
Nomenclature
Chapter 7

A  Aspect ratio or area of main heater
a  Accommodation coefficient
\( \bar{c} \)  Average velocity of molecules (ms\(^{-1}\))
c\(_p\)  Specific heat at constant pressure (J Kg\(^{-1}\) °C\(^{-1}\))
c\(_v\)  Specific heat at constant volume (J Kg\(^{-1}\) °C\(^{-1}\))
d  Molecular diameter (m)
D\(_h\)  Hydraulic diameter (m)
g  Acceleration of gravity (ms\(^{-2}\))
Gr  Grashof number
h  Combined heat transfer coefficient from absorber to cover (Wm\(^{-2}\) °C\(^{-1}\))
h'  Heat transfer coefficient of material of known conductivity (Wm\(^{-2}\) °C\(^{-1}\))
h\(_b\)  Heat transfer coefficient for flow across panel wall (Wm\(^{-2}\) °C\(^{-1}\))
h\(_c\)  Heat transfer coefficient for flow across the inside of the panel due to convection and conduction (Wm\(^{-2}\) °C\(^{-1}\))
h\(_p\)  Heat transfer coefficient for flow across panel (Wm\(^{-2}\) °C\(^{-1}\))
h\(_r\)  Heat transfer coefficient for flow across the inside of the panel due to radiation (Wm\(^{-2}\) °C\(^{-1}\))
h\(_s\)  Heat transfer coefficient for flow across standard insulation (Wm\(^{-2}\) °C\(^{-1}\))
k  Thermal conductivity (Wm\(^{-1}\) °C\(^{-1}\))
L  Linear dimension (m)
m  Wall molecule mass (Kg)
m'  Gas molecule mass (Kg)
M  Mass of one mole (kg mol\(^{-1}\))
N\(_A\)  Avogadro's number
Nu  Nusselt number
p  Gas pressure (Nm\(^{-2}\))
P\(_C\)  Critical pressure when Ra = Ra\(_C\)
Pr  Prandtl number
q  Power dissipated in central heater (W)
\( Q \) 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)

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

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

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

\( \gamma \) \( c_p/c_v \)

\( \Delta \theta \) 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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TABLE 2.1 Energy input by fuel and sector in Petajoules for U.K. low grade heat needs (\(\leq 80^\circ C\)) for 1976 and 2025 as predicted by Leach [1]

<table>
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## TABLE 2.3 Basic Prometheus configuration to heat 100 houses

### Store

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<td>Width</td>
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<td>Height</td>
<td>4 m</td>
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<td>Volume</td>
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<td>Storage material pebbles, density</td>
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<td>Specific heat capacity</td>
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<td>Store insulation; thermal conductivity</td>
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### Collector

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<td>Optical efficiency averaged over useful incident angles (Ta)</td>
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<td>Item</td>
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<td>168</td>
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*Refer to Chapter 2.*
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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<tr>
<td>( R_h/£ \text{ yr}^{-1} )</td>
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\[ i=0.05 \quad f=0.04 \quad 6600 \quad 6000 \quad 6300 \]

\[ PV_{Ch} \quad i=0 \quad f=0.04 \quad 8500 \quad 17800 \quad 20200 \]

\[ i=0 \quad f=0.02 \quad 7500 \quad 11700 \quad 12500 \]
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<td>Cost of collectors £1980/m²</td>
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<td>64</td>
<td>64</td>
<td>72</td>
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<tr>
<td>Cost of store £1980/m³</td>
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<td>Collector area/Storage volume (m²/m³)</td>
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<td>0.36</td>
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<td>Total system capital cost £1980</td>
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<td>659000</td>
<td>1740000</td>
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Collector area required to heat type A5 house (27.5 GJ/annum)/m²

Storage volume required for type A5 house /m³

Cost per A5 house/£1980

Chapter 2 reference numbers
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<tr>
<th>Store temperature rise/°C</th>
<th>Cost/£1982 per KWh recovered energy seasonal storage</th>
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<td>Pit storage</td>
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<td>Storage in clay</td>
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<td>Multiple well systems in rock</td>
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<td>Aquifers</td>
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<th>Name of Store/ or Centre of Study</th>
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<th>Design Study Material</th>
<th>Cost Per House Supplied by System</th>
<th>% of Annual House Heating</th>
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<td>Constructed</td>
<td>Water</td>
<td>19 320</td>
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<td>Studsvik, Sweden</td>
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## TABLE 3.1  Thermal Characteristics of Basic Type AO House

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<th>Component</th>
<th>Area A (m²)</th>
<th>U-value (Wm⁻²°C⁻¹)</th>
<th>UA (W°C⁻¹)</th>
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<td>Month</td>
<td>Days in month</td>
<td>Solar radiation on a South-facing vertical surface (KWh/m²/month)</td>
<td>Solar radiation on a South-facing surface 30° to horizontal (KWh/m²/month)</td>
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<tr>
<td>House type</td>
<td>Insulation level</td>
<td>Total house specific loss (W°C⁻¹)</td>
<td>Net annual space and water heating demand (GJ)</td>
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<td>A0</td>
<td>Basic (1975 Building Regs.)</td>
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<td>A1 + 50 mm loft insulation (100 mm total)</td>
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<td>A2 + fill cavity with fibre</td>
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<td>A4 + extra layer of glazing (i.e. double)</td>
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<tr>
<td>A6</td>
<td>A5 + cavity increased to 100 mm</td>
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<td>A7</td>
<td>A6 + 25 mm floor edge insulation</td>
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<td>A7 + all windows on south side</td>
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<td>A11</td>
<td>A10 + cavity increased to 200 mm</td>
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TABLE 3.4 Thermal characteristics of Basic Type BO house

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<th>Component</th>
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<td>Total fabric specific loss</td>
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<td>192 W°C⁻¹</td>
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<td>Total house specific loss</td>
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TABLE 3.5 Thermal Characteristics of existing houses with different levels of retrofitted insulation.

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<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>
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<td>Basic (average UK housing stock)</td>
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<td>B2</td>
<td>B1 + fibre-fill cavity (50 mm)</td>
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<td>B3</td>
<td>B2 + 50 mm of loft insulation (150 mm total)</td>
<td>215</td>
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<td>B4</td>
<td>B3 + extra layer of glazing (i.e. double)</td>
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<td>B5 + 100 mm external wall insulation</td>
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*Table 4.1: Art collector, test facilitators, and reported systems in the United Kingdom*
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**Column Index**

1. Time (hrs : min)
2. Mass flow rate (kg hr⁻¹)
3. Total insolation (Wm⁻²)
4. Air temperature rise passing through collector (Tᵢ - Tᵢ₋₁)/(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. (Tᵢ - Tₛ)/(Tᵢ₋₁)
12. (Tᵢ₋₁ - Tₛ)/Tᵢ
### TABLE 5.2(a)  Results of steady state testing on D.C.Hall collector

<table>
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<tr>
<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>Air mass</td>
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<td>Air temp. at condenser</td>
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Results of steady state testing of structured polyanionate collector.

**TABLE 5.2(b)**
TABLE 5.3 Collector configuration modelled for transient analysis by RRDCT.

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<tr>
<th>Property</th>
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<td>Collector length (along flow)</td>
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<td>Collector width</td>
<td>1.00 m</td>
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<tr>
<td>Cover to plate spacing</td>
<td>0.05 m</td>
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<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>
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<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>
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<td>Plate absorbtance</td>
<td>0.95 at $\theta = 0$ falling slightly as $\theta$ increases</td>
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<td>Emissivity of upper surface of the plate (diffuse)</td>
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<td>Emissivity of duct surface (diffuse)</td>
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<td>Emissivity of cover (diffuse)</td>
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<td>Cover polycarbonate thinkness</td>
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<td>Mass flow rate</td>
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<td>Thickness of plate and of duct-back</td>
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TABLE 5.4 Results of transient and steady state testing with multi node model

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<th>Transient 0.5mm (DY2)</th>
<th>Transient 2mm (DY4)</th>
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<td>$N$</td>
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<td>$\tau_{C}/(\text{min})$</td>
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<tr>
<td>$F_{R_1U}/(\text{W m}^{-2} \text{K}^{-1})$</td>
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<td>$F_{R_2\alpha}$</td>
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<td>$K F_{R_3\alpha}$</td>
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<td>$\hat{\sigma} F_{R_5\alpha}$</td>
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</table>

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

$* = \text{at low fluid inlet temperatures}$
<table>
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<th>( F_R(t_0), k_n )</th>
<th>( \sigma_{F_R(t_0)} k_n )</th>
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**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

**TABLE F.4**

\( n = 1 \) \( k_n = 1 \)

\( F = .482116948686 \)

\( \text{ETAO}= .513943004957 \) \( U = 7.89668509365 \)

**DATA SETS ACCEPTED FOR ANALYSIS SO**

\( n/F^* = .344370221325 \) \( T^* = .003 \)

\( .379994726798 | .022 |
| .39186542197 | .022 |
| .427499407851 | .021 |
| .415619232633 | .022 |
| .427494067851 | .021 |
| .43968903069 | .021 |
| .474993407823 | .021 |
| .463118575305 | .023 |
| .451243732827 | .024 |
| .427494067851 | .025 |
| .415619232633 | .024 |
| .427494067851 | .022 |
| .43968903069 | .021 |
| .415619232633 | .021 |
| .106875216963 | .062 |
| .9374170905E-2 | .057 |
| .8.3123846526E-2 | .053 |
| .8.3123846526E-2 | .054 |
| .7.1249011306E-2 | .025 |
| .7.1249011306E-2 | .046 |
| .3.56245056542E-2 | .044 |
| .11.874352181 | .024 |
| .201872198707 | .019 |
| .33249536106 | .029 |
| .24937153958 | .049 |
| .213747033926 | .056 |
| .213747033926 | .061 |
| .166247693054 | .067 |
| .178122528271 | .069 |
| .5.9374170905E-2 | .073 |
| .3.56245056542E-2 | .067 |
| .3.56245056542E-2 | .059 |
| .3.56245056542E-2 | .052 |
| .7.1249011306E-2 | .044 |
| .7.1249011306E-2 | .025 |
| .130623187399 | .02 |
| .3749704361 | .044 |
| .6.6247693053 | .053 |
| .130623187399 | .052 |
| .106873516493 | .028 |
| .94372857875 | .02 |

**POINTS ON THERMAL PERFORMANCE CHARACTERISTIC BO**

FROM LEAST SQUARES FITS EACH WAY

MINIMUM ETA= .235453187816

MAXIMUM ETA= .714184616622

\( U = 7.33893217894 \)

\( U = 13.9616808148 \)
### TABLE 5.8
Temperature distribution within DY1 collector (0.2mm thick plate and duct back) during ASHRAE steady state testing, $T_a = 293k$, $I = 700\, \text{Wm}^{-2}$, Wind = 1m s$^{-1}$, $T_{sky} = 273k$

<table>
<thead>
<tr>
<th>$T_1/k$</th>
<th>$T_e/k$</th>
<th>$\bar{T}_p/k$</th>
<th>$\bar{T}_b/k$</th>
<th>$T_m/k$</th>
<th>$F_{RLU}$ (Wm$^{-2}$ $^\circ$C$^{-1}$)</th>
<th>$\eta$</th>
<th>$F_{aveLU}$ (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>
<td>2.762</td>
<td>.645</td>
<td>3.111</td>
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<tr>
<td>343</td>
<td>364.98</td>
<td>365.28</td>
<td>357.16</td>
<td>354.00</td>
<td>2.902</td>
<td>.476</td>
<td>3.230</td>
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<tr>
<td>383</td>
<td>396.47</td>
<td>396.94</td>
<td>391.47</td>
<td>389.73</td>
<td>3.044</td>
<td>.293</td>
<td>3.362</td>
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<td>423</td>
<td>427.23</td>
<td>428.06</td>
<td>425.00</td>
<td>425.11</td>
<td>3.185</td>
<td>.095</td>
<td>3.503</td>
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<tr>
<td>433</td>
<td>435.13</td>
<td>435.94</td>
<td>433.57</td>
<td>434.06</td>
<td>3.226</td>
<td>.037</td>
<td>3.564</td>
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</table>

### 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 = 1m s$^{-1}$, $T_{sky} = 273k$

<table>
<thead>
<tr>
<th>$T_1/k$</th>
<th>$T_e/k$</th>
<th>$\bar{T}_p/k$</th>
<th>$\bar{T}_b/k$</th>
<th>$T_m/k$</th>
<th>Energy lost per unit time per unit area W m$^{-2}$</th>
<th>$F_{RLU}$ (Wm$^{-2}$ $^\circ$C$^{-1}$)</th>
<th>$F_{aveLU}$ (Wm$^{-2}$ $^\circ$C$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>303</td>
<td>300.34</td>
<td>300.41</td>
<td>301.23</td>
<td>301.67</td>
<td>40.34</td>
<td>4.034</td>
<td>4.653</td>
</tr>
<tr>
<td>343</td>
<td>333.32</td>
<td>333.79</td>
<td>336.20</td>
<td>338.16</td>
<td>146.66</td>
<td>2.932</td>
<td>3.247</td>
</tr>
<tr>
<td>383</td>
<td>365.41</td>
<td>366.41</td>
<td>370.42</td>
<td>374.20</td>
<td>266.50</td>
<td>2.961</td>
<td>3.282</td>
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<tr>
<td>423</td>
<td>396.74</td>
<td>398.43</td>
<td>403.88</td>
<td>409.87</td>
<td>397.80</td>
<td>3.060</td>
<td>3.404</td>
</tr>
<tr>
<td>433</td>
<td>404.46</td>
<td>406.34</td>
<td>412.12</td>
<td>418.73</td>
<td>432.40</td>
<td>3.088</td>
<td>3.439</td>
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<tr>
<td>*303</td>
<td>301.62</td>
<td>301.71</td>
<td>302.03</td>
<td>302.31</td>
<td>20.98</td>
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<td>2.035</td>
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<td>*433</td>
<td>405.92</td>
<td>407.78</td>
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<td>419.46</td>
<td>410.30</td>
<td>2.93</td>
<td>3.245</td>
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</table>

* $T_{sky} = 293k$
<table>
<thead>
<tr>
<th>Structured Polycarbonate Collector</th>
<th>Indoor</th>
<th>Transient</th>
<th>Steady State</th>
<th>ASHRAE</th>
<th>D.C. Hall Collector</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (°C)</td>
<td>0.4 - 3.0°C</td>
<td>7.5°C</td>
<td>7.0°C</td>
<td>9.2°C</td>
<td>7.46°C</td>
</tr>
<tr>
<td>Relative Humidity (%)</td>
<td>60%</td>
<td>71%</td>
<td>63%</td>
<td>58%</td>
<td>73%</td>
</tr>
<tr>
<td>Wind Speed (m/s)</td>
<td>2</td>
<td>1.5</td>
<td>1.0</td>
<td>0.7</td>
<td>0.6</td>
</tr>
<tr>
<td>Test Method</td>
<td>Ta</td>
<td>TW</td>
<td>TR</td>
<td>F (m³/h)</td>
<td>P (W)</td>
</tr>
<tr>
<td>Summary of Collector Testing Results</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Material</td>
<td>Reflective index (n)</td>
<td>Solar (0.2-4.0µm)</td>
<td>Infrared (3.0-500µm)</td>
<td>Expansion coefficient (°C⁻¹)</td>
<td>Temperature Limits (°C)</td>
</tr>
<tr>
<td>--------------------------------</td>
<td>----------------------</td>
<td>-------------------</td>
<td>----------------------</td>
<td>-----------------------------</td>
<td>-------------------------</td>
</tr>
<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>
</tr>
<tr>
<td>Plexiglass (Acrylic)</td>
<td>1.49</td>
<td>125 mil</td>
<td>125 mil</td>
<td>8.29 x 10⁻⁵</td>
<td>80-90</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>
</tr>
<tr>
<td>Tedlar P.V.F. (fluorocarbon)</td>
<td>1.46</td>
<td>4 mil</td>
<td>4 mil</td>
<td>5.95 x 10⁻⁵</td>
<td>110-170</td>
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<tr>
<td>Mylar (Polyester)</td>
<td>1.64-1.67</td>
<td>5 mil</td>
<td>5 mil</td>
<td>2.00 x 10⁻⁵</td>
<td>150-200</td>
</tr>
<tr>
<td>Sunlite (Fibre glass)</td>
<td>1.54</td>
<td>25 mil</td>
<td>25 mil</td>
<td>2.98 x 10⁻⁵</td>
<td>95-100</td>
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<tr>
<td>Float glass (Glass)</td>
<td>1.518</td>
<td>125 mil</td>
<td>125 mil</td>
<td>10.21 x 10⁻⁶</td>
<td>230</td>
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<tr>
<td>Temper glass (Glass)</td>
<td>1.518</td>
<td>125 mil</td>
<td>125 mil</td>
<td>10.21 x 10⁻⁶</td>
<td>230-250</td>
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<tr>
<td>Clear limesheet glass (Low iron glass)</td>
<td>1.51</td>
<td>125 mil</td>
<td>125 mil</td>
<td>10.64 x 10⁻⁶</td>
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</tr>
<tr>
<td>Clear lime temper glass (Low iron glass)</td>
<td>1.51</td>
<td>125 mil</td>
<td>125 mil</td>
<td>10.64 x 10⁻⁶</td>
<td>200</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>
</tr>
</tbody>
</table>

Source: Gary, H.P. 'Treatise on solar energy' Vol.1, A Wiley Interscience Publication, Chichester, 1982
<table>
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<td>Sydney</td>
<td>Skyson/MPD</td>
<td>In-house</td>
<td>DBU</td>
<td>Sympatec/Granes</td>
<td>Phillips</td>
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<tr>
<td>Material</td>
<td>Most</td>
<td>Copper/Glass</td>
<td>Stainless Steel</td>
<td>Copper</td>
<td>Copper</td>
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<tr>
<td>Source</td>
<td>Helios 14, Cardiff University</td>
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</table>
TABLE 6.3  Key to collector variable features, used to obtain Figure 6.19

<table>
<thead>
<tr>
<th>Cover Material:</th>
<th>Feature</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>cover 1</td>
<td>plate glass, thickness</td>
<td>6.0 mm</td>
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</tr>
<tr>
<td>cover 2</td>
<td>polycarbonate, thickness</td>
<td>2.0 mm</td>
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</table>

<table>
<thead>
<tr>
<th>Thickness of the plate and of the duct-back:</th>
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<tbody>
<tr>
<td>DY1</td>
</tr>
<tr>
<td>DY2</td>
</tr>
<tr>
<td>DY3</td>
</tr>
<tr>
<td>DY4</td>
</tr>
<tr>
<td>DY5</td>
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</table>

<table>
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<th>Air Flow in the Rear-Duct:</th>
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</thead>
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<td>flow 1</td>
</tr>
<tr>
<td>flow 2</td>
</tr>
<tr>
<td>flow 3</td>
</tr>
<tr>
<td>Gas</td>
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<td>Air</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>$h_p/(Wm^{-1}°C^{-1})$</th>
<th>$Q_A/(Wm^{-2})$</th>
<th>$T_f/°C$</th>
<th>$h_f/(Wm^{-2}°C^{-1})$</th>
<th>$h_c/(Wm^{-2}°C^{-1})$</th>
<th>$ΔT/°C$</th>
</tr>
</thead>
<tbody>
<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>
<td>0.704</td>
</tr>
<tr>
<td></td>
<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>
</tr>
<tr>
<td></td>
<td>10.1</td>
<td>21.3</td>
<td>1.725</td>
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<td>11.07</td>
<td>20.33</td>
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<td>1.915</td>
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<tr>
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<td>33.3</td>
<td>2.195</td>
<td>50.05</td>
<td>13.00</td>
<td>30.80</td>
<td>0.180</td>
<td>2.632</td>
</tr>
<tr>
<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>
<td>1.720</td>
</tr>
<tr>
<td></td>
<td>10.35</td>
<td>38.8</td>
<td>1.621</td>
<td>46.12</td>
<td>12.66</td>
<td>36.49</td>
<td>0.185</td>
<td>1.750</td>
</tr>
<tr>
<td>Air, p = 81 torr</td>
<td>10.3</td>
<td>43</td>
<td>1.567</td>
<td>51.24</td>
<td>12.86</td>
<td>40.44</td>
<td>0.189</td>
<td>1.669</td>
</tr>
<tr>
<td>Air, p = 71 torr</td>
<td>10.2</td>
<td>24.9</td>
<td>0.925</td>
<td>13.60</td>
<td>10.88</td>
<td>24.22</td>
<td>0.172</td>
<td>0.847</td>
</tr>
<tr>
<td>Freon/Air</td>
<td>10.3</td>
<td>22.1</td>
<td>1.685</td>
<td>19.88</td>
<td>11.29</td>
<td>21.11</td>
<td>0.170</td>
<td>1.856</td>
</tr>
<tr>
<td></td>
<td>10.1</td>
<td>17.8</td>
<td>1.635</td>
<td>12.59</td>
<td>10.73</td>
<td>17.17</td>
<td>0.166</td>
<td>1.789</td>
</tr>
<tr>
<td>Carbon Tet/Air</td>
<td>10.1</td>
<td>17.9</td>
<td>1.645</td>
<td>12.83</td>
<td>10.74</td>
<td>17.26</td>
<td>0.166</td>
<td>1.803</td>
</tr>
<tr>
<td></td>
<td>10.4</td>
<td>27.9</td>
<td>1.986</td>
<td>34.75</td>
<td>12.14</td>
<td>26.16</td>
<td>0.175</td>
<td>2.303</td>
</tr>
<tr>
<td></td>
<td>10.5</td>
<td>31.3</td>
<td>2.081</td>
<td>43.28</td>
<td>12.66</td>
<td>29.14</td>
<td>0.178</td>
<td>2.450</td>
</tr>
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<td></td>
<td>10.6</td>
<td>34.9</td>
<td>2.461</td>
<td>59.80</td>
<td>13.59</td>
<td>33.91</td>
<td>0.182</td>
<td>3.082</td>
</tr>
<tr>
<td></td>
<td>10.6</td>
<td>36.7</td>
<td>2.245</td>
<td>58.59</td>
<td>13.53</td>
<td>33.77</td>
<td>0.183</td>
<td>2.712</td>
</tr>
<tr>
<td>Air P = 0.3 torr</td>
<td>10.2</td>
<td>16.6</td>
<td>0.547</td>
<td>3.504</td>
<td>10.38</td>
<td>16.42</td>
<td>0.165</td>
<td>0.414</td>
</tr>
<tr>
<td>Air P = 0.35 torr</td>
<td>10.9</td>
<td>45.7</td>
<td>1.135</td>
<td>39.51</td>
<td>12.87</td>
<td>43.72</td>
<td>0.193</td>
<td>1.088</td>
</tr>
<tr>
<td>Air P = 16 torr and changing</td>
<td>11.2</td>
<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>
</tr>
</tbody>
</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

Dwelling mean weekly hot water consumption m³

Total No. of dwellings: 87
Overall mean weekly consumption: 0.841 m³/week
Standard deviation: 0.351 m³/week

Contribution due to OAPs flats
(10/2 occupants)

FIGURE 2.5 MEAN WEEKLY HOT WATER CONSUMPTION FOR 87: VARIOUS SITES.
FROM 'THE PERFORMANCE OF DOMESTIC WET HEATING SYSTEMS' BY G.A. RICKUP.[7]

SOLAR m=0
SOLAR m=2
BLACKBODY AT 40°C

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 INSOLATION ON A HORIZONTAL SURFACE.

FIGURE 2.8
AVERAGE GLOBAL SOLAR RADIATION ON A HORIZONTAL SURFACE

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

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)

APARTMENT BUILDING (INTENSE POPULATED AREAS)  (LARGE SCALE)

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 2.15 PROTO-PROMETHEUS
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 2.18 PROTO-PROMETHEUS TEMPERATURE DISTRIBUTION (WITH FAN ON), ON 22ND SEPTEMBER 1981 AT 14:25 HRS.
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 3.8 cm
STANDARD DEVIATION 0.95 cm
**FIGURE 2.21** PROTO-PROMETHEUS STORE TEMPERATURE, FROM 25th SEPTEMBER 1981 TO 2nd OCTOBER 1981 UNDER STANSTATION (FAN OFF).

**FIGURE 2.22** ENERGY DEMAND FOR A 3-BEDROOM HOUSE BUILT TO A75 BUILDING REGULATIONS (TYPE A1) 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 A1 house.

**Figure 2.24** Effect of changing the collector area on the % of annual energy supplied by Prometheus to a type A1 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 m$^2$ PER HOUSE AND 28 m$^2$ OF COLLECTOR PER HOUSE) FOR A TYPE 1 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 AS HOUSE.
**Figure 2.29** Design of Costed Prometheus to provide 100 houses with 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 A0 House

Figure 3.2 Net space heating demand for Type A0, 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 M75 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 7.12 and 24m² 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 80 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
The curves correspond to the following relations:

- **McAdams**  
  \[ h_w = 5.7 + 3.5V \]

- **Watmuff**  
  \[ h_w = 2.8 + 3.0V \]

- **Lloyd**  
  \[ h_w = \frac{0.15 \times R_0^{0.4} \times 2.6 \times L}{L+W} \]  
  for \( T_a=10^\circ C, T_e=15^\circ C, L=1m, W=1m \).

- **Sparrow**  
  \[ h_w = \frac{k 	imes 0.86 \times R_0^{0.4} \times T_e^{0.6}}{2 \times L} \]  
  for \( T_a=10^\circ C, T_e=15^\circ C, L=0.1m \).

- **Green**  
  \[ h_w = \frac{(h_0 + h_f^{15})^{0.8}}{2 \times L/W} \]  
  for \( A=1.4m^2, 45^\circ \) inclination.

- **KIND**  
  For combined length \( 2.4m \), width \( 1.2m \)  
  height \( 4.5m \), \( T_a = 25^\circ C \).

**Figure 4.7** Correlations for Wind Heat Loss Coefficient
**Figure 4.8** Flow diagram of 'EFFIC2' (See Appendix B) a program to calculate the efficiency of a PTC air heating collector.
INPUT
ENVIRONMENTAL PARAMETERS $I, V, T_a$

COLLECTOR CONFIGURATION ($n$) $\theta_1, \theta_2, k, h, L, W, D$
$V_e, x$

COLLECTOR VARIABLES $T_c, m$

INITIAL ESTIMATE OF $T_r, T_m$

CALCULATE
$R_c$
$N_u$
$h_1, h_2$
$h_r$
$U_b$
$U_e$
$U_L$
$F_r$
$F_e$
$Q_u$

SEE EQUATION 4.25
4.23
4.22
4.27
4.4
4.15
4.16
4.20
4.19
4.18

$\eta = \frac{Q_u}{A_l}$

CALCULATE NEW ABSORBER TEMPERATURE
$T_{p_{\text{new}}} = T_c + \frac{(Q_u/A_l)(1 - F_r)}{ULF_r}$

IS

$T_F = T_{F_{\text{new}}}$

FLOW DIAGRAM OF 'EFFIC' (SEE APPENDIX B) A PROGRAM TO CALCULATE THE EFFICIENCY OF A TOP DUCT WARMING COLLECTOR
FIGURE 4.10 RESPONSE OF ZERO AND LONG TIME CONSTANT COLLECTOR TO CHANGING INSOLATION
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 NNW 1966-1975)
Figure 5.2  D.C. Hall Collector
FIGURE 5.3 ANGULAR VARIATION OF TRANSMITTANCE OF 2mm THICK POLYCARBONATE (REFRACTIVE INDEX = 1.526, EXTINCTION COEFFICIENT = 20 m⁻¹)

FIGURE 5.4 TEE-PIECES USED FOR ABSORBER FINS IN D.S. 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-walled 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.
**Figure 5.11**  Response of structured polycarbonate collector to a step change in insolation from 750 W/m² to zero with a fluid flow rate of 7.2 kg/h.**

**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 0.75° (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 $F_{U,V}$, 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 FLOW FLOWS

FIGURE 5.21 SAMPLE OUTPUT OF D.C. WALL 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 $F'_{CO_2}$ ON THE DIFFUSE FRACTION FOR A SINGLE-GLAZED FLAT-PLATE COLLECTOR. SOURCE: POROSKI [34].
Figure 5.29 Computer generated steady state and transient efficiency curve for 0.5 mm absorber plate.
FIGURE 5.30  TRANIENT 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 $F_{n L}$, $F_{e}(t)$, AND $\delta F_{e L}$ 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 3P COLLECTOR ON 14/6/93-15/6/93.
**Figure 5.30** Standard error in $F_{\text{ul}}$ versus $N$, the number of previous time steps influencing the collector's present performance under transient conditions for the structured polycarbonate collector.

**Figure 5.31** Estimated efficiency curve from $F^{\text{het}}$ and $F^{\text{ul}}$.

**Figure 5.32** Least squares fit, maximum and minimum.

**Figure 5.33** 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 ABSTRACT). 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 Wm⁻², AVERAGE INTENSITY 2.11 Wm⁻², STANDARD DEVIATION ± 0.9 Wm⁻².

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

Measured heat loss with collector operating under stagnation and assuming $(P/e) = 0.72$ plotted against average air velocity parallel to collector plane and measured 5mm above collector plane.

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 DC hull collector with non-selective absorber (Nextel). 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.

ASHRAC steady state

+ ZERO TESTING $T_{sky} = T_c - 20$

- $T_{sky} = T_m$
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 (303 K).

**Figure 5.55** Collector temperature profile for model collector under steady state and zero testing conditions for the same mean absorber plate temperature (366 K).
**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** $F_{\text{in,1}}$ versus mean fluid temperature for collector dy1 under zero testing and ashape 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 P-T CHALL TYPE COLLECTOR (MAXIMUM ABSORBER)
Figure 5.61: Steady state efficiency of solar collector (black chrome) measured during operation and indoor testing. Source: Taylor, P.J. "Performance of selecting and non-selective solar thermal absorbers in a working installation." Solar World Congress ed by S.V. Szovary, Vol. 2, pp 1149 - 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.

Figure 6.5  Maximum improvement to flat plate collector performance by increasing the absorptivity α and the emittance ε.
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-SELECTIVE SOLAR THERMAL ABSORBERS IN A WORKING INSTALLATION." SOLAR WORLD CONGRESS ED BY S. N. SENDEROGLU, VOL 2, PP 1149-1153.
Figure 6.8
Economy curves for different methods of heat loss reduction.

Figure 6.9
**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. $I_{in} = 211\, \text{W/m}^2$, $T_a = 28^\circ\text{C}$, $T_{in} > T_a$, $T_c = T_a$ and air velocity = 1.5 m/s.
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 DIFFERENT SIZES Z, AND TWO LEVELS OF INCIDENT INSOLATION.
FIGURE 6.14  EFFICIENCY CURVE FOR A COMBINED PARABOLIC CONCENTRATOR COMPARED WITH A FLAT PLATE COLLECTOR. SOURCE: ARCONAAT NATIONAL LABORATORY TECH REPORT.

FIGURE 6.15  GLOBAL AND DIFFUSE INSOLATION MONTH BY MONTH AT 45° SOUTH/FACING 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
- **TA** = 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 4 and Figure 3).  

(i) $S\bar{J}/T\!A\!J$, flow 2  
(ii) $S\!O\!M/T\!A\!M$, flow 2  
(iii) $S\!O\!D/T\!A\!D1$, flow 2  
(iv) $S\!O\!M/T\!A\!M$, flow 3  
(v) $S\!1\!M/T\!A\!M$, flow 2  
(vi) $S\!O\!D/T\!A\!D1$, flow 3  
(vii) $S\!1\!D1/T\!A\!D1$, flow 2  
(viii) $S\!O\!D/T\!A\!D2$, flow 3  
(ix) $S\!1\!D2/T\!A\!D1$, flow 2  
(x) $S\!1\!D3/T\!A\!D1$, flow 2  
(xi) $S\!1\!D/T\!A\!D1$, flow 2.
**FIGURE 6.20** 'FMTC' AIR HEATING SOLAR COLLECTOR DEVELOPED BY G.E. [42]

Figure 6.22: Instantaneous efficiencies of the FMC 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 VISCOITY 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 7.6**  SCHEMATIC DEPICTING EFFECT OF GAP SPACING ON CONDUCTANCE
Figure 7.2: Plot of $h_c$ versus plate separation $s$, $T_{wall} < 10^\circ C$, $T_{air} = 325^\circ C$, $P = 1.4$ mbar.

Figure 7.8: $h_c$ versus tilt angle to the horizontal for air with $T_{wall} = 10^\circ C$, for various absorber temperatures ($T_a$).
Figure 7.9

Heat transfer coefficients 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, separated 5 cm, cold plate temperature 5°C, hot plate 30°C.
FIGURE 7.12

Heat transfer coefficient for gases of different molecular weight. For $S = 5$ cm, cold plate temperature $10^\circ C$, hot plate temperature $30^\circ C$. 
FIGURE 7.13  
COST VERSUS HEAT TRANSFER COEFFICIENT FOR DIFFERENT GASES.  
$\bar{s} = 5$ cm, VOLUME OF GAS REQUIRED FOR EACH SQUARE METRE OF COLLECTOR IS 50 LITRES.
FIGURE 7.14 VARIATION OF HEAT TRANSFER COEFFICIENT $h_c$ WITH PRESSURE FOR A FLAT PLATE COLLECTOR, $S = 5$ cm, $T = 293 K$, $T_s = 823 K$ FOR CURVE 1, 273 K FOR CURVE 2 AND 473 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: HOLMTRUP A AND GARG H.P. 'MINIMIZING CONVECTIVE HEAT LOSSES'. SOLAR ENERGY VOL 25, NO 6, 7523.

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. LEGGE, PERGAMON PRESS 1979
FIGURE 7.19  
HEAT TRANSFER COEFFICIENT $h_c$ DUE TO NATURAL CONVECTION FOR AIR AT ATMOSPHERIC PRESSURE BETWEEN TWO PARALLEL FLAT PLATES SPACING $5cm$, $T_r = 293K$, WITH A HONEYCOMB PAD WITH SLATS, ASPECT RATIO 5
Figure 7.20: Thermal Conductivity versus Rayleigh Number for various gases. $T_i = 10^\circ C$, $T_e = 80^\circ C$, $\delta = 5 \text{ cm}$. 
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

$S = 5 \text{ cm}; \ T_c = 10^\circ \text{C}$
Figure 7.23
Guard Ring Heater

Figure 7.24
Guard Ring Unbalance versus Measured Heat Transfer across a 5cm thick Styrofoam 69/9 Sample
FIGURE 7.25 ACYRILC TEST PANEL

FIGURE 7.26 SCHEMATIC DIAGRAM OF GUARDED HOT PLATE APPARATUS
Figure 7.27 Copper Cold Plates.
**Figure 7.28** Measured and Theoretical Heat Transfer Coefficients for Different Gases Between Two Parallel Plates, s = 5 cm, Various Temperature Difference.
Figure 7.29 Theoretical and measured heat transfer $h_c$ for air and argon.

- Experimental points:
  - X Argon
  - O Air
  - Argon with helium

Diagram shows the variation of heat transfer coefficient $h_c$ with hot plate temperature $T_2$.
**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 RGR INSULATION 3. FAN MOTOR 4. S. 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 7.2
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 PRINT "IN CASE 1"
30 SHORT DEMAND(12)!
40 SHORT SOL(12,24)
50 ASSIGN 1 TO "SUN DATA"
60 READ# 1 SOL();
70 SHORT TEM(12,24)
80 ASSIGN 2 TO "TEM DATA"
90 READ# 2 TEM();
100 DIM MONTHS(12)(13)
110 ASSIGN 3 TO "MONTH"
120 READ# 3 ➔ MONTHS()
130 SHORT DAYS(12)
140 ASSIGN 4 TO "DAYS"
150 READ# 4 ➔ DAYS()
160 PRINT USING 200
170 TOTSUN=0
180 PRINT "TOTSUN=TOTAL ANNUAL SOLAR RADIATION"
190 PRINT ""
200 IMAGE ///"""" SOLAR RADIATION AT KNOX DISTRIBUTION OF HOUINGLY GLOBAL IRRIGATION ///""
210 PRINT USING 220
220 PRINT "IMAGE ///"""" ON A HORIZONTAL SURFACE IN MJ/m2 ///""
230 FOR M=1 TO 12 !
240 PRINT TAB (6&M):MONTHS(M)!!
250 NEXT M
260 FOR H=1 TO 24 !
270 FOR M=1 TO 12 !
280 PRINT TAB (M&6):SOL(M,H)!!
290 TOTSUN=TOTSUN+SOL(M,H)!!DAYS(M) ! calculate total annual solar radiation.
300 NEXT M
310 PRINT ""!
320 PRINT ""!
330 PRINT "TOTAL ANNUAL SOLAR RADIATION = "TOTSUN"MJ/m2"
340 REM **************************** DATA INPUT ****************************
350 FI=.9 ! HEAT TRANSFER FACTOR
360 Cea=.837 ! SPECIFIC HEAT OF STORE MATERIAL (KJ/KgC)
370 MEX=PEBBLES ! STORAGE MATERIAL
380 WIDTH=10 ! STORAGE WIDTH IN METERS
390 HEIGHT=4 ! STORAGE HEIGHT IN METERS
400 LENGTH=280 ! STORAGE LENGTH IN METERS
410 HOUSE=100 ! NUMBER OF HOUSES SURVED BY STORE
420 DENSITY=1600 ! DENSITY OF STORAGE MATERIAL (Kg/m3)
430 H=2 ! OVERALL COLLECTOR HEAT LOSS COEFFICIENT (W/m2)
440 COLAREA=2800 ! TOTAL AREA OF COLLECTORS SURVING STORE (m2)
450 COND=.036 ! THERMAL CONDUCTIVITY OF STORAGE INSULATING MATERIAL (W/m2C)
460 THICK=.6 ! THICKNESS OF INSULATING MATERIAL (m)
470 T=8 ! OPTICAL EFFICIENCY AVERAGED OVER USEFUL INCIDENT ANGLES
480 YEARS=1 ! NUMBER OF YEARS PROGRAM TO RUN DO NOT USE MORE THAN 1 IF qaux=0
490 T=10 ! TEMPERATURE OF GROUND SURROUNDING STORE (C)
500 T=50=30 ! MINIMUM STORAGE TEMPERATURE (C)
510 REM **************************** STORE ****************************
520 REM **************************** CONSTANTS ****************************
530 REM (K) ! READS MONTHLY DATA OF HEATING LOAD FOR EACH HOUSE (MJ)
1000 REM ******************************************************************************
1010 PRINT TAB (10); "MONTHS()"; "="; DEMAND(); "! print heating demand each month"
1020 TOTD=TOTD+DEMAND(); "calculate total annual heating demand"
1030 DEMAND(); " heating demand per m2 of collector"
1040 CL
1050 PRINT "TOTAL ENER$countPER ANNUUM"; "(TOTD/1000)"; "GJ"; "="; "TOTDI"
1060 W=KHW
1070 PRINT " "
1070 PRINT USING 1080
1080 IMAGE " "
1090 IMAGE " "
1100 IMAGE " "
1110 PRINT "IT = Threshold Level (collector will only operate above this internal temperature) (W/m2)"
1120 PRINT "C = Temperature Averaged over periods of collector operation (C)
1130 PRINT "T = Threshold Level (collector will only operate above this temperature (C/m2)
1140 PRINT "n = Normalized Net Heat to Storage =q(t-1-1) (C/m2)
1150 PRINT "qT = Useful Heat Collected= qN+1m (C/m2)
1160 PRINT "qM = Normalized Total Monthly Load (MJ/m2)
1170 PRINT "qL = Normalized Total Monthly Storage (MJ/m2)
1180 PRINT "qAux= qN+4aux (C/m2)
1190 PRINT "Tillis= qN+qM+qL+qAux (C/m2)
1200 PRINT "qY= qM+qL+qAux (C/m2)
1210 PRINT "qAux = aux
1220 PRINT " "
1230 FOR i=1 TO 12
1240 TEMPERATURE = 0
1250 TSOL=0
1260 TEMP=1
1270 FOR j=1 TO 24
1280 IT=IT+1/(TSOL-TEM(i,j))
1290 IF IT=TSOL(J,1)/.0056 THEN GOTO 1330 'test if radiation level is above average ambient temp and is on each day'
1300 PRINT "TEMP=TEMP+TEM(i,j)"
1310 TSOL=TSOL+i+1+1
1320 NEXT J
1330 NEXT I
1340 IF TIM=0 THEN GOTO 1360 'prevent dividing by zero'
1350 TEMP=TEMP/TIM
1360 TIM=TIM+3600/DAYS(i)
1370 TSOL=TSOL+1
1380 IF TIM=0 AND COUNT=0 THEN GOTO 1770
1390 t=3600/DAYS(i)+1000000
1400 qT=Fi=TSOL-ULP1(TSO-TEMP)";
1410 G=USAS(TSO-Tq)=ETC
1420 qN=q(T-Demand)-I-1]"; (Fi=ULP1(USAS(TSO-Tq)-ETC))
1430 qM=TSOL=TSOL/ETC
1440 If TSOL=THEN GOTO 1530
1450 T=S
1460 IF Ts=30 THEN GOTO 1530
1470 TEMP=TEMP/2
1480 IF COUNT=0 THEN GOTO 1770
1490 IF NUT=1 THEN GOTO 1520
1500 EXTRAS(TSO-Ts)=c*m
1510 NUT=1
1520 GOTO 1540
1530 Tsn=(TSD+Tst)
1540 qN=TSN-TEM/Y]
1550 qT=q(T-Demand)-I-1]"; (Fi=ULP1(USAS(TSO-Tq)-ETC))
1560 qM=TSOL=TSOL/ETC
1570 qT=qN/T-TATE
1580 COUNT=1
1590 qAux=qN+EXTRA
1600 EXTRA=0
1610 IF TST=THEN qAux=0
1620 PRINT USING 1960 ; MONTHS(i), IT, TSOL, TEMP, TT, TSOL, qT, TATE, qN, qAux
1630 X=i+10
1640 Y=TIM/2
1650 PLOT X, Y
1660 TOTAL=TOTAL+TSOL
1670 qT=qT+qT
1680 qT=qT=qnt+qN
1690 qT=qT=qst+qTT
1700 qT=qT=qst+qtt
1710 qT=qT=qst+qtt
1720 qT=qT=qst+qtt
1730 qT=qT=qst+qtt
1740 qT=qT=qst+qtt
1750 qT=qT=qst+qtt
1760 qT=qT=qst+qtt
1770 qT=qT=qst+qtt
1780 qT=qT=qst+qtt
1790 qT=qT=qst+qtt
1800 PRINT " "
1810 PRINT USING 1970 ; "TOTAL", qnt, qst, TT, TATE, qAux
1820 PRINT " "
1830 SUN=1-(qAux/TOTAL)
1840 PRINT USING 1980 ; "% OF ENERGY SUPPLIED BY SOLAR SYSTEM"
1850 PRINT USING 1990 ; "% OF TOTAL ENERGY COLLECTED ABOVE THRESHOLD";
1860 PRINT USING 2000 ; "% OF SOLAR ENERGY COLLECTED";
1870 PRINT " "
1880 PRINT " "
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, 3500, 980, 770, 770, 770, 770, 770, 5270, 7450
2020 END
### SOLAR-RADIATION-AT-NEW-DISTRIBUTION OF HOURLY-GLOBAL IRADIATION-------

ON A HORIZONTAL SURFACE IN MJ/m²

<table>
<thead>
<tr>
<th>JAN</th>
<th>FEB</th>
<th>MAR</th>
<th>APR</th>
<th>MAY</th>
<th>JUN</th>
<th>JUL</th>
<th>AUG</th>
<th>SEP</th>
<th>OCT</th>
<th>NOV</th>
<th>DEC</th>
<th>TOTAL ANNUAL SOLAR RADIATION</th>
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<td>0</td>
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<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>3410.94 MJ/m²</td>
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**STORE**

**LENGTH** = 280 Meters  **WIDTH** = 10 Meters  **HEIGHT** = 4 Meters

**UP/NET** = 11200 m³

**STORAGE MATERIAL** = PEBBLES  **DENSITY** = 1600 kg/m³  **SPECIFIC HEAT** = 0.837 KJ/kg°C

**STORE INSULATION THICKNESS** = .6 m  **THERMAL CONDUCTIVITY** = .056 W/m°C

**COLLECTOR**

**TOTAL COLLECTOR AREA** = 2800 m²

**Fi** = HEAT TRANSFER FACTOR (equivalent to Fr heat removal factor if store has a good heat exchanger) = .9

**UL** = OVERALL HEAT LOSS COEFFICIENT = 1

**Ta** = OPTICAL 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

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<tr>
<th>JAN</th>
<th>FEB</th>
<th>MAR</th>
<th>APR</th>
<th>MAY</th>
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<tr>
<td>7750</td>
<td>770</td>
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<td>770</td>
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**AUG** = 770  **SEP** = 770  **OCT** = 1790  **NOV** = 3270  **DEC** = 7450

**TOTAL ENERGY DEMAND OF HOUSE PER ANNUM** = 41.69 GJ (11580.555556 kWh)

**SYSTEM OPERATION**

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<tr>
<th>ITH</th>
<th>Tso</th>
<th>tF</th>
<th>tT</th>
<th>qN</th>
<th>tsf</th>
<th>qT</th>
<th>Tm</th>
<th>Tm</th>
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<td>APR 29.00</td>
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<td>10.296</td>
<td>350.7</td>
<td>88.0</td>
<td>46.4</td>
<td>206.67</td>
<td>118.57</td>
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<tr>
<td>MAY 25.13</td>
<td>46</td>
<td>10.562</td>
<td>477.1</td>
<td>209.5</td>
<td>85.5</td>
<td>243.97</td>
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<td>25.4</td>
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<td>10.728</td>
<td>549.9</td>
<td>182.1</td>
<td>121.2</td>
<td>219.16</td>
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<td>10.674</td>
<td>513.1</td>
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<td>143.08</td>
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<td>10.562</td>
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<td>10.116</td>
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<td>110.0</td>
<td>125.0</td>
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<td>63.93</td>
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<td>8.2</td>
<td>171.0</td>
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<td>141.99</td>
<td>198.57</td>
<td>9.1</td>
<td>-57.0</td>
</tr>
</tbody>
</table>

**TOTAL** = 462.00  1026.80  1488.93  390.2 -462.0

**% 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** = 1299.0  7849.90  3594.33518 03 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.
DECLARE PROGRAM TOPAIR

DECLARE THE TOP LOSS COEFFICIENT FOR A SINGLE GLASS FORM

DECLARE SEE DUFFIE & BECKMAN P204

DECLARE REM

DECLARE PROGRAM TOPAIR

DECLARE FOR I=0 TO 20

DECLARE 60 TP=10+I*5 ; ABSORBER TEMP

DECLARE 70 TA=10 ; Ambient temp (C)

DECLARE 80 WIND=1 ; Wind speed (ms-1)

DECLARE 90 EP=.95 ; Absorber emissivity

DECLARE 100 EC=0.8 ; Cover plate emissivity

DECLARE 110 S=5 ; Plate separation (cm)

DECLARE 120 G=9.812 ; Acceleration due to gravity (ms-2) at LONDON

DECLARE 130 K=0.0257 ; Thermal conductivity of gas at Tave (W/m·C)

DECLARE 140 B=0 ; Tilt angle=0(Horizontal)

DECLARE 150 CP=1007 ; Heat capacity of air (J/KgK)

DECLARE 155 EP=1007 ; Heat capacity of GAS BETWEEN COVER AND ABSORBER KJ/m

DECLARE 160 S=5/100 ; CONVERT TO METERS

DECLARE 170 L=1

DECLARE 180 KH=1

DECLARE 190 SW=29LW/(L+W)

DECLARE 200 REM

DECLARE 210 TC=TA+(TP-TA)/2 ; guess the cover temp

DECLARE 220 TI=273.15/TC ; CONVERT TO KELVIN

DECLARE 230 TA=273.15/TA ; CONVERT TO KELVIN

DECLARE 240 TC=273.15/TC ; CONVERT TO KELVIN

DECLARE 250 TP=273.15/TP ; CONVERT TO KELVIN

DECLARE 260 T=TP+TC ; CONVERT TO KELVIN

DECLARE 270 DT2-T1 ; TEMP DIFFERENCE DELTA T

DECLARE 280 Tave=TI+DT2/2 ; AVERAGE GAS TEMPERATURE

DECLARE 290 DEN=352.91/Tave

DECLARE 300 h=Tave/10.000076+0.0034406

DECLARE 310 VIS=Tave/10.0000464+0.000046351

DECLARE 320 VOL=1/Tave ; THERMAL VOLUME EXPANSION COEFFICIENT ONLY HOLDS FOR PERFECT

DECLARE GAS

DECLARE 330 V=VIS/DEN ; KINEMATIC VISCOSITY

DECLARE 340 Gr=G*VIS/3*DT/V/2 ; GRASHOF NUMBER

DECLARE 350 Pr=CP/VIS/K

DECLARE 360 Ra=Gr*Pr ; RAYLEIGH No

DECLARE 370 REM

DECLARE 380 N=1-1708/(R*CO2+B)/5830;1/(1+3-1)

DECLARE 410 IF N<0 THEN N=0 ; TAKE ONLY POSITIVE TERMS

DECLARE 420 N=N-1708/(R*CO2+B); N2 ; NUSUILT No

DECLARE 430 h=4*N;1-4444N/5*(1-SIN(1.808)+1.61708/(R*CO2+B))+N2 ; NUSUILT No

DECLARE 440 h=0.0000000067*(TP+2+TC)+1/EP/1/EC-1 ; RAD FROM PLATE TO COVER PRINT

DECLARE 450 hsky=0.0000000067*EC*(TC+2+TA)+1/TC+TA ; RAD COVER TO SKY

DECLARE 470 DT2+TC=TA

DECLARE 480 Tave=TA+DT/2

DECLARE 490 DEN=352.91/Tave

DECLARE 500 KH=Tave/10.000076+0.0034406

DECLARE 510 VIS=Tave/10.0000464+0.000046351

DECLARE TOP 500 PRINT " TOP LOSS COEFFICIENT (see Duffie & Beckman pp204) Pr. TOP 

DECLARE TOP 640 PRINT " COLLECTOR CHARACTERISTICS 

DECLARE 650 PRINT " ABSORBER TEMP=1;TP;K(" TP=273.15;C")

DECLARE 685 IF I>0 THEN GOTO 10

DECLARE 690 PRINT " WIND SPEED=" WIND/1/2.01 THEN TC=TC ELSE GOTO 630

DECLARE 700 PRINT " "

DECLARE 710 PRINT " "

DECLARE 720 PRINT " "

DECLARE 730 PRINT " "

DECLARE 740 PRINT " "

DECLARE 750 PRINT " "

DECLARE 760 PRINT " "

DECLARE 770 PRINT " "

DECLARE 780 PRINT " "

DECLARE 790 PRINT " "

DECLARE 800 PRINT " "

DECLARE 810 PRINT " "

DECLARE 820 PRINT " "

DECLARE 830 PRINT " "

DECLARE 840 PRINT " "

DECLARE 850 PRINT " "

DECLARE 860 PRINT " "

DECLARE 870 PRINT " "

DECLARE 880 PRINT " "

DECLARE 890 PRINT " "

DECLARE 900 PRINT " "

DECLARE 905 PRINT " "

DECLARE 910 PRINT " "

DECLARE 915 NEXT I

DECLARE 920 END
APPENDIX C

THE EFFECT OF THERMAL CAPACITANCES ON THE PERFORMANCE OF SOLAR COLLECTORS

Barrie W. Jones and Tadj Oreszczyn
The Open University, Milton Keynes MK7 6AA, UK

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

DY1-5 plate and duct-back thicknesses (5)
f(θ) transmittance - absorbance function of the collector
FR collector heat-removal factor
HPA(I) heat-transfer coefficient plate (or duct-back) to air in the I'th segment of the duct
M duct air flow rate
NI number of duct segments
PON threshold power for switch on of air flow
s irradiance in cover plane
S0 solar beam irradiance
S1 diffuse irradiance on a horizontal surface
SP irradiance absorbed by plate
TA ambient temperature
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>TI</td>
<td>air inlet temperature</td>
</tr>
<tr>
<td>TK</td>
<td>sky temperature</td>
</tr>
<tr>
<td>TO</td>
<td>air outlet temperature</td>
</tr>
<tr>
<td>UT</td>
<td>overall U-value of the collector</td>
</tr>
<tr>
<td>W</td>
<td>width of collector</td>
</tr>
<tr>
<td>WIND</td>
<td>wind speed</td>
</tr>
<tr>
<td>M, J, D, D1, D2, D3</td>
<td>various subscripts (see text)</td>
</tr>
<tr>
<td>η</td>
<td>steady-state thermal efficiency of the collector</td>
</tr>
<tr>
<td>ηn</td>
<td>daily averaged thermal efficiency of the collector</td>
</tr>
<tr>
<td>θ</td>
<td>angle between collector normal and solar beam</td>
</tr>
</tbody>
</table>

1 INTRODUCTION

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 {I}, {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

| collector length (along flow) | 4.00 m |
| collector width (W)           | 1.00 m |
| cover to plate spacing        | 0.05 m |
| rear duct gap                 | 0.01 m |
| back insulation               | dry glass fibre, thickness 0.10 m |
| edge insulation               | dry glass fibre, thickness 0.05 m |
| material of plate and duct-back| duralumin HS15TB |
| cover                         | polycarbonate, thickness 2.00 mm |
| plate absorbtance             | 0.95 at θ=0, falling slightly as θ increases |
| emissivity of upper surface of the plate (diffuse) | 0.10 |
| emissivity of duct surfaces (diffuse) | 0.91 |
| emissivity of the cover (diffuse) | 0.85 |
| thermal properties of air at 283 K for ambient air, at 303 K elsewhere | |
| latitude                      | 52°N |
| collector tilt (to horizontal)| 35° |
| collector orientation         | south-facing |
| thickness of plate and of duct-back | |
| DY1                            | 0.2 mm |
| DY2                            | 0.5 mm |
| DY3                            | 1.0 mm |
| DY4                            | 2.0 mm |
| DY5                            | 5.0 mm |
| collector time-constant (flow 1) | |
| flow 0                         | |
| flow 1                         | all TI M = 0.0600 kg s⁻¹ (PON irrelevant) |
| flow 2                         | TI = 303 K M = 0.0600 kg s⁻¹ PON = 128 W |
| flow 3                         | TI = 323 K M = 0.0562 kg s⁻¹ PON = 124 W |

Air flow in the rear-duct

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 \( T_i \) the value of \( M \) will be different from \( 0.0600 \) kg s\(^{-1}\): at \( T_i = 323 \) K, \( M = 0.0562 \) 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 PON is slightly less. The values of PON are shown in Table 1. Note that the values of PON are for a 4 m \( \times \) 1 m collector, and not for the whole array. These values of PON 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)

\[
\eta = F_R \cdot f(\theta) - U_L (T_i - T_A)/S
\]

where \( f(\theta) \) is such that

\[
SP = f(\theta) \cdot S
\]

For the steady state efficiency curve \( S \) is beam irradiance normal to the cover, such that \( S = 700 \) W m\(^{-2}\). Furthermore, \( T_A = 293 \) K, \( T_K = 273 \) K, \( \text{WIND} = 1.0 \) m s\(^{-1}\), \( M = 0.0600 \) 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 \( T_i \) 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 \((T_i-T_A)/S\) this is \(-2.83 \) W m\(^{-2}\) K\(^{-1}\), giving a value of \( U_L \) of 3.44 W m\(^{-2}\) K\(^{-1}\).

The value of \( F_R \cdot U_L \) increases as \( T_i \) 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 rearduct 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 TA varies sinusoidally through the day (Figure 3(a)) with an amplitude of 5 K. Note that there are two temperature curves for 21 December, TAD1 and TAD2. 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 insolations, the prefix S0 denoting the clear sky irradiance normal to the beam, and the prefix S1 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 TA, and in overcast conditions it is 10 K below TA. 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

\[
\bar{n} = \text{total energy extracted by the air flow in the day} / \text{integration of } S \text{ over the day.}
\]

(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 \(\bar{n}\) is never being wrongly evaluated.

In order to plot \(\bar{n}\) on Figure 2 it is necessary to re-define the abscissa \((\text{TI}-\text{TA})/S\). TI is constant (303 K or 323 K), and for TA 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 \((\text{TI}-\text{TA})/S\) 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 \(\bar{n}\) 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 \((\text{TI}-\text{TA})/S\)). 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 TA and TK do not make an appreciable contribution to the spread of \(\bar{n}\) with thermal capacitance on the scale of Figure 2.

The advantage of low mass could, in principle, be more marked under intermittent insolation. 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 {1}.

Figure 2 also shows that the values of \( \bar{\eta} \) 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 \( \bar{\phi} \) is always zero, thus raising \( f(\bar{\phi}) \), 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 \( \bar{\eta} \), the daily average of \( \bar{T}_0 \) 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 \( \bar{T}_0 \) 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 \( \bar{T}_0 \). However the variation of \( \bar{\eta} \) 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 \( (\bar{T}_I-\bar{T}_A)/\bar{S} \) 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.
Figure 2  Steady-state efficiency ($\eta_\text{ steady}$ - 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) SUD/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) SLD1/TAD1, flow 2  (viii) SOD/TAD2, flow 3  (ix) SLD2/TAD1, flow 2  (x) SLD3/TAD1, flow 2  (xi) SLD/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.
214
940 NEXT K
950 NEXT K
960 ZE=SOR (ZEYY/(NF-NC))
970 PRINT "ETAQ":E,"*"/'":ZE
980 UN=X(NC)
990 PRINT "FU"="U","-"/"":Z(NC)
1000 PRINT "TABLE F.4"
1010 FOR K=1 TO N
1020 C(K)=X(1)/E
1030 PRINT K,C(K)
1035 NEXT K
1040 F=U/(HLOG (1-U/H))
1050 PRINT "F"=IF
1060 E=E/F
1070 U=U/F
1080 PRINT "ETAD"="E,"=U;
1090 PRINT "DATA SETS ACCEPTED FOR ANALYSIS"=NF
1100 REM READ DATA TO GENERATE THERMAL PERFORMANCE CURVE
1110 ASSIGN 1 TO "TRANSD700"
1120 NF=0
1130 READ# 1 : I,X(NK),Y,T(NK)
1140 IF I=0 AND X(NK)=0 THEN GOTO 1570
1150 I=I+1
1160 FOR K=2 TO NK
1160 L=NF-K+1
1170 READ# 1 : I,X(L),Y,T(L)
1175 IF I=0 AND X(L)=0 THEN GOTO 1570
1180 IF I#1 THEN GOTO 1130
1190 L=L+1
1200 NEXT K
1210 GOTO 1400
1220 FOR K=2 TO NK
1220 L=NF-K+2
1230 X(L)=X(L-1)
1240 T(L)=T(L-1)
1250 NEXT K
1260 READ# 1 : I,X(I),Y,T(I)
1270 IF I=0 AND X(I)=0 THEN GOTO 1570
1275 IF I#1 THEN GOTO 1150
1280 IF I#1 THEN GOTO 1150
1290 I=I+1
1300 E=0
1310 X(NC)=0
1320 FOR K=1 TO NK
1330 E=E+X(K)*C(K)
1340 X(NC)=X(NC)+T(K)
1350 NEXT K
1360 Y=Y/(FRE)
1370 X(NC)=X(NC)/NF*E
1380 PRINT Y,X(NC)
1390 REM CALC LEAST SOR TO THERMAL PERFORMANCE
1400 SX=SSX+X(NC)
1410 SY=SSY+Y
1420 SYI=SSX+X(NC)*Y(NC)
1430 SYI=SSY+Y
1440 SXI=SSX+X(NC)*Y(NC)