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APPENDIX A

ELECTRICITY PRICE

This appendix shows the electricity price rates for households based on the announcement of rates by the Metropolitan Power Board which was effective from 1st August 1981.

Households Electricity Price Rate per kWh	
Kilowatt-hour	Price Rate (Baht)/kWh
1-5	5.00 (for first five unit)
6-15	0.70
16-25	0.90
26-35	1.17
36-100	1.79
101-150	1.89
151-300	1.97
301-400	2.03
401-	2.10

APPENDIX B

MATHEMATICAL DESCRIPTION

This appendix lists the mathematical equations for each system component. Generally any consistent system of units can be used, except where specified otherwise.

<u>Pressure Relief Valve</u>

Let

T_{max} : boiling point of the liquid at maximum allowable pressure P_{max}

T_1, T_2 : inlet and outlet temperatures, respectively

$\dot{M}_1$, $\dot{M}_2$: inlet and outlet mass flow rates, respectively

C_p : heat capacity of the liquid

$\dot{Q}_{discard}$: rate of energy discarded due to boiling

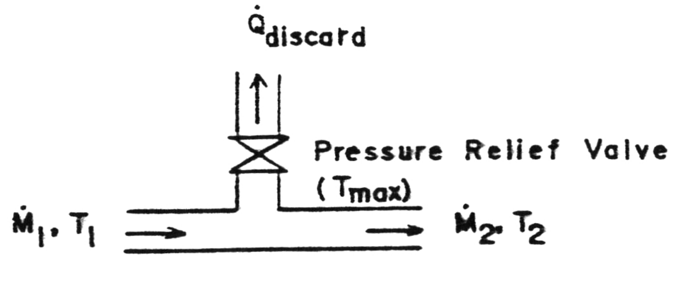


FIGURE B.1.1 PRESSURE RELIEF VALVE

If $T_1 \leq T_{max}$ or $\dot{M}_1 = 0$, then

$$\dot{M}_2 = \dot{M}_1$$

$$T_2 = T_1$$

$$\dot{Q}_{discard} = 0$$

If $T_1 > T_{max}$ and $\dot{M}_1 > 0$, then

$$\dot{M}_2 = \dot{M}_1$$

$$T_2 = T_{max}$$

$$\dot{Q}_{discard} = \dot{M}_1 C_p (T_1 - T_{max})$$

On/off Auxiliary Heater

Let

T_{set} : the temperature desired of the flowstream

$\dot{Q}_{max}$: the maximum heat delivery rate of the auxiliary heater

$\dot{Q}_{aux}$: the rate of auxiliary heat actually supplied

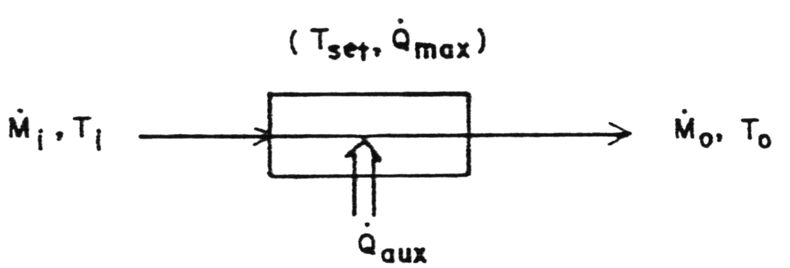


FIGURE B.2.1 ON/OF AUXILIARY HEATER

If $\dot{M}_i = 0$ or $T_i \geq T_{set}$, then

$$\dot{M}_o = \dot{M}_i$$

$$T_o = T_i$$

$$\dot{Q}_{aux} = 0$$

Otherwise,

$$\text{If} \quad \dot{Q}_{max} > \dot{M}_i C_p (T_{set} - T_i) \begin{cases} \dot{M}_o = \dot{M}_i \\ T_o = T_{set} \\ \dot{Q}_{aux} = \dot{M}_i C_p (T_{set} - T_i) \end{cases}$$

or,

$$\text{If} \quad \dot{Q}_{max} \leq \dot{M}_i C_p (T_{set} - T_i) \begin{cases} \dot{M}_o = \dot{M}_i \\ T_{set} = T_i + \dfrac{\dot{Q}_{max}}{\dot{M}_i C_p} \\ \dot{Q}_{aux} = \dot{Q}_{max} \end{cases}$$

<u>Stratified Liquid Storage Tank</u>

Let

M_i : mass of liquid in segment i

T_i : temperature of liquid in segment i

T_{env} : temperature of tank surroundings

UA_i : product of overall heat transfer coefficient and tank surface area for segment i

N : number of equal volume, fully mixed (uniform temperature) tank segments

$\dot{Q}_{loss}$: rate of heat loss to the tank surroundings

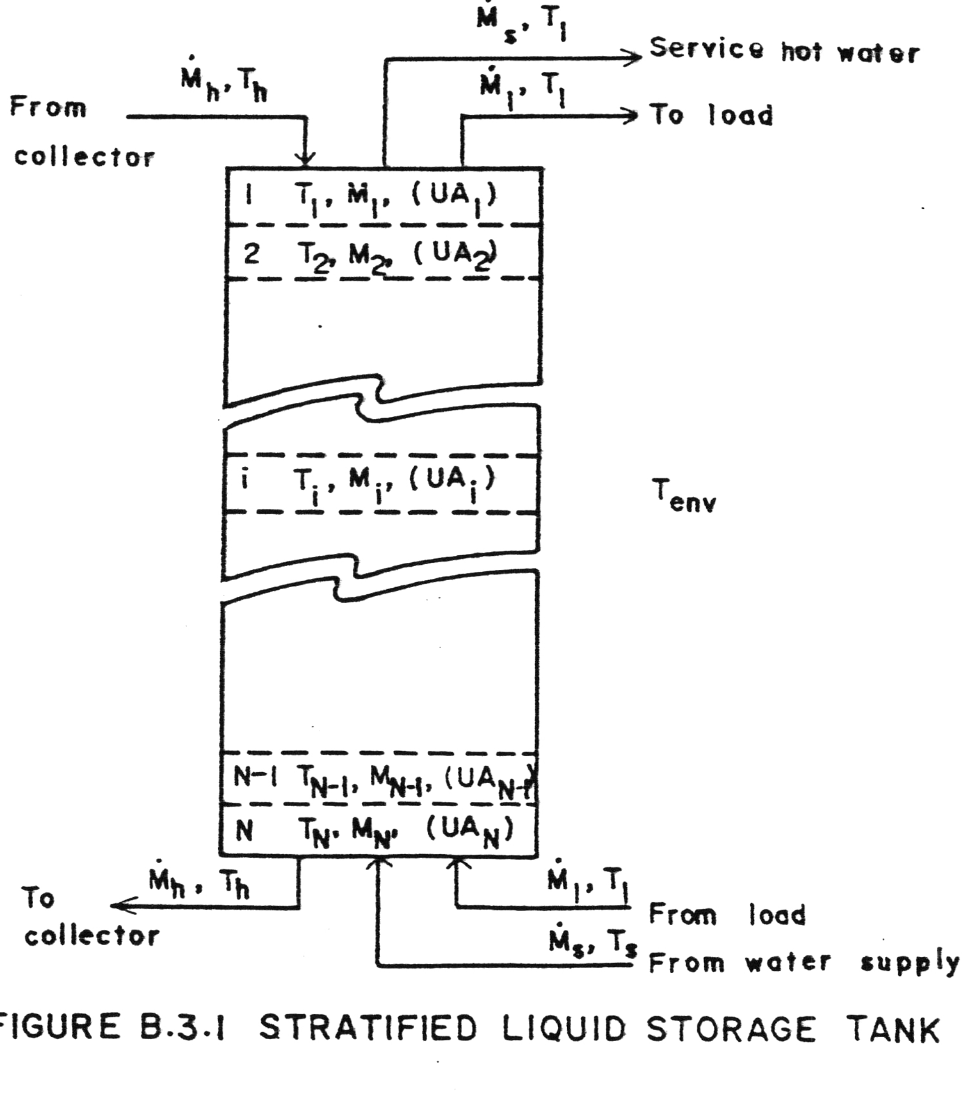


FIGURE B.3.1 STRATIFIED LIQUID STORAGE TANK

1) N = 1 (No stratification)

$$M_1 C_p \frac{dT_1}{dt} = \dot{M}_h C_p (T_h - T_1) + \dot{M}_l C_p (T_l - T_1) + \dot{M}_s C_p (T_s - T_1)$$

$$- UA_1 (T_1 - T_{env})$$

2) $N \geq 2$ (Temperature stratification)

$$M_1 C_p \frac{dT_1}{dt} = \dot{M}_h C_p T_h - \dot{M}_l C_p T_1 - \dot{M}_s C_p T_1 + \gamma \dot{M}_{net} C_p T_2$$

$$- (1 - \gamma) \dot{M}_{net} C_p T_1 - UA_1 (T_1 - T_{env})$$

$$M_N C_p \frac{dT_N}{dt} = \dot{M}_l C_p T_l + \dot{M}_s C_p T_s - \dot{M}_h C_p T_N - \gamma \dot{M}_{net} C_p T_N$$

$$+ (1 - \gamma) \dot{M}_{net} C_p T_{N-1} - UA_N (T_N - T_{env})$$

For $N \geq 3$ and $2 \leq i \leq N-1$,

$$M_i C_p \frac{dT_i}{dt} = - \dot{M}_{net} C_p T_i + \gamma \dot{M}_{net} C_p T_{i+1} + (1 - \gamma) \dot{M}_{net} C_p T_{i-1}$$

$$- UA_i (T_i - T_{env})$$

where

$$\dot{M}_{net} = | \dot{M}_l + \dot{M}_s - \dot{M}_h |$$

$$\gamma = \begin{cases} 1 & \text{if } \dot{M}_l + \dot{M}_s \geq \dot{M}_h \\ 0 & \text{otherwise} \end{cases}$$

For all values of N,

$$\dot{Q}_{loss} = \sum_{i=1}^{N} UA_i (T_i - T_{env})$$

<u>Flow Diverter</u>

Let

γ : an input control function ($0 \leq \gamma \leq 1$)

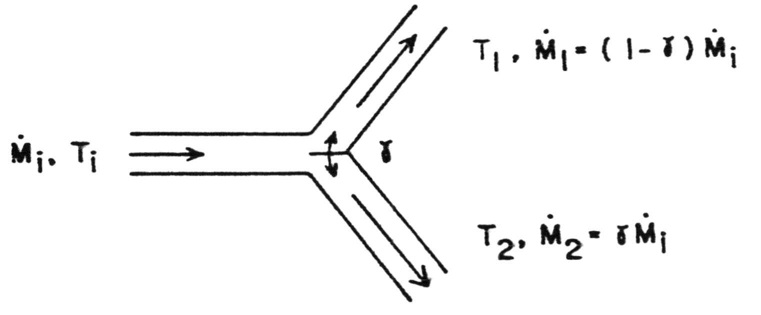


FIGURE B.4.1 FLOW DIVERTER

For $0 \leq \gamma \leq 1$,

$$T_1 = T_2 = T_i$$

$$\dot{M}_1 = (1 - \gamma)\dot{M}_i$$

$$\dot{M}_2 = \gamma \dot{M}_i$$

<u>Flow Mixer</u>

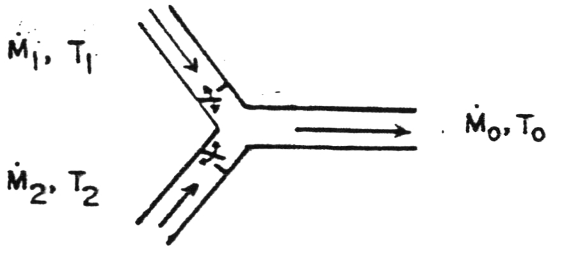


FIGURE B.5.1 FLOW MIXER

For $\dot{M}_1 > 0$ or $\dot{M}_2 > 0$,

$$\dot{M}_o = \dot{M}_1 + \dot{M}_2$$

$$T_o = (\dot{M}_1 T_1 + \dot{M}_2 T_2) / \dot{M}_o$$

Otherwise,

$$\dot{M}_1 = \dot{M}_2 = \dot{M}_o = 0$$

and T_o can be set at an arbitrary value, such as $(T_1 + T_2) / 2$

<u>Pump and Fan</u>

Let

$\dot{M}_{max}$: maximum mass flow rate

γ : input control function ($0 = \gamma = 1$)

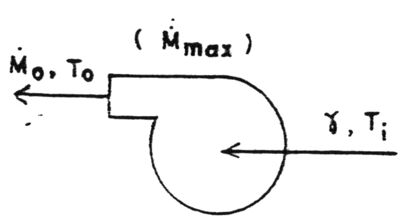


FIGURE B.6.1 PUMP AND FAN

$$\dot{M}_o = \gamma \dot{M}_{max}$$

$$T_o = T_i$$

Absorption Air-Conditioner

Let

$\dot{Q}_{input}$: rate of input energy required for space cooling

$\dot{Q}_{cool}$: rate of heat removal by space cooling

$\dot{Q}_{reject}$: rate of heat rejection to the atmosphere by the cooling tower

$\dot{M}_i$, $\dot{M}_o$: inlet and outlet hot water mass flow rates, respectively

T_i, T_o : inlet and outlet hot water temperatures, respectively

T_{con} : cooling water inlet temperature

γ : control function ($\gamma = 0$ or 1)

$\dot{M}_{max}$: maximum mass flow rate of hot water pump

$\dot{Q}_{ref}$: nominal input energy requirement rate

($\dot{Q}_{ref} = 52{,}750$ kJ/hr for a 3-ton unit)

$\dot{Q}_{cap}$: nominal cooling capacity

($\dot{Q}_{cap} = 37{,}980$ kJ/hr for a 3-ton unit)

C_p : heat capacity of hot water

T_{wb} : ambient wet bulb temperature

If $\gamma = 0$ (air-conditioner is off), then

$$\dot{Q}_{input} = \dot{Q}_{cool} = \dot{Q}_{reject} = 0$$

$$\dot{M}_i = \dot{M}_o = 0$$

$$T_o = T_i$$

If $\gamma = 1$ (air-conditioner is on), then

$$\dot{M}_o = \dot{M}_i = \gamma\dot{M}_{max}$$

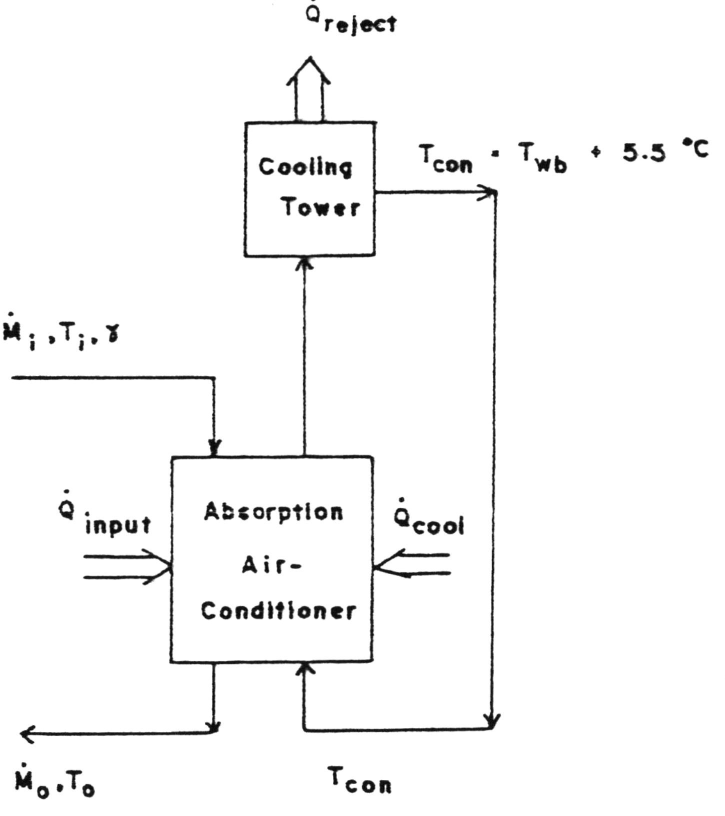


FIGURE B.7.1 ABSORPTION AIR-CONDITIONER

COOLING CAPACITY DATA

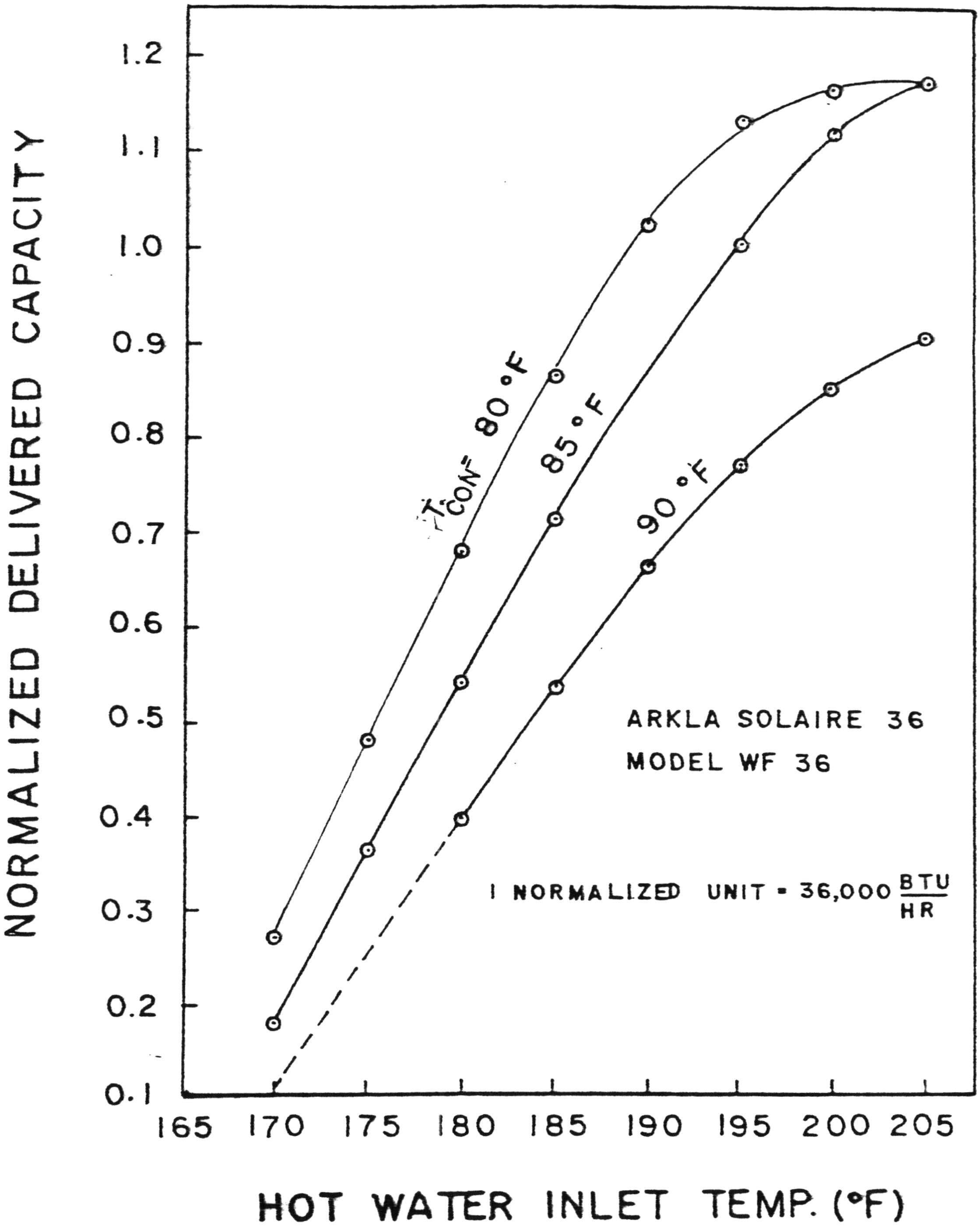


FIGURE B.7.2 NORMALIZED COOLING CAPACITY FUNCTION, f_1

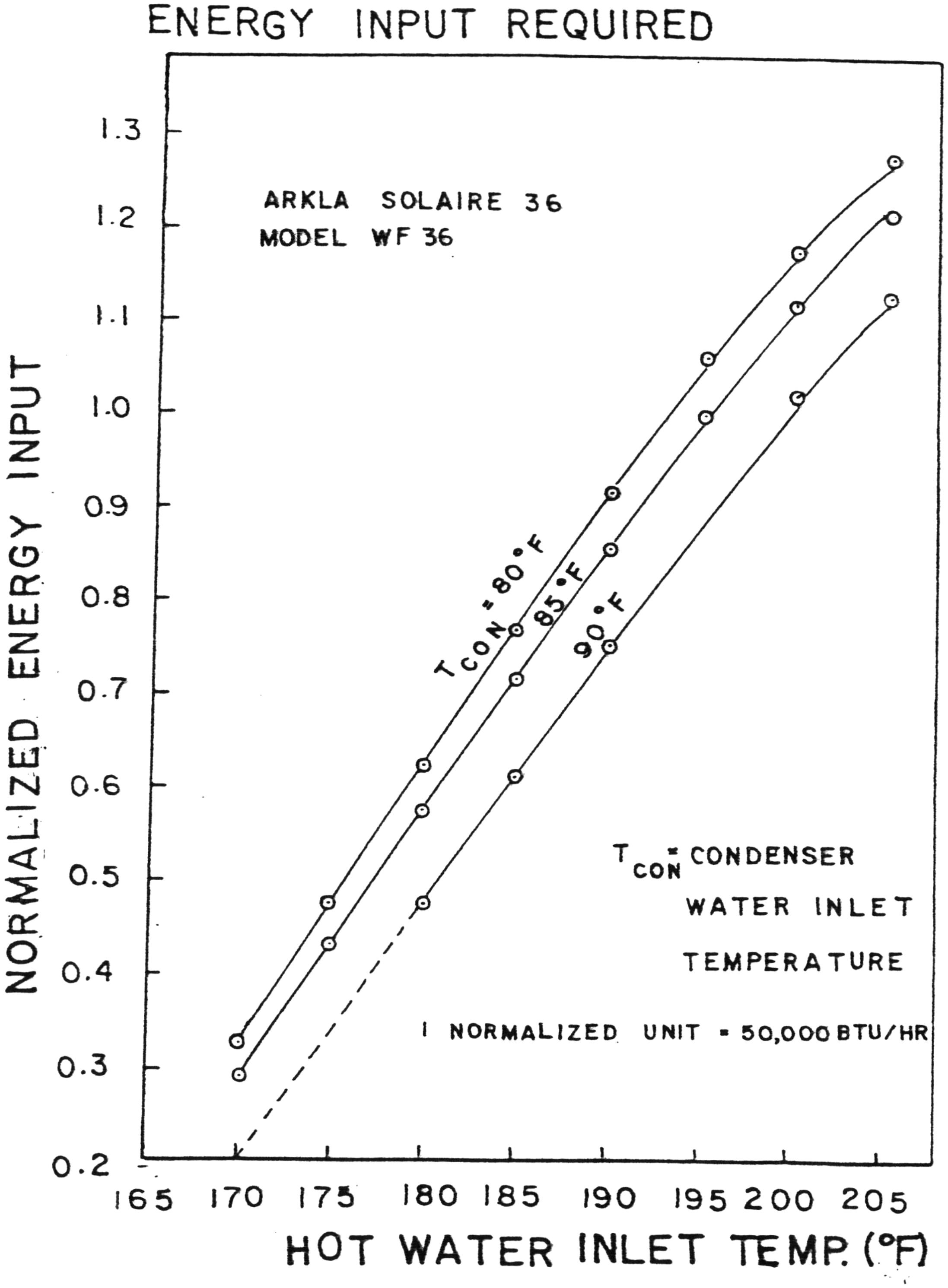


FIGURE B-7-3 NORMALIZED INPUT ENERGY REQUIREMENT RATE FUNCTION, f_2

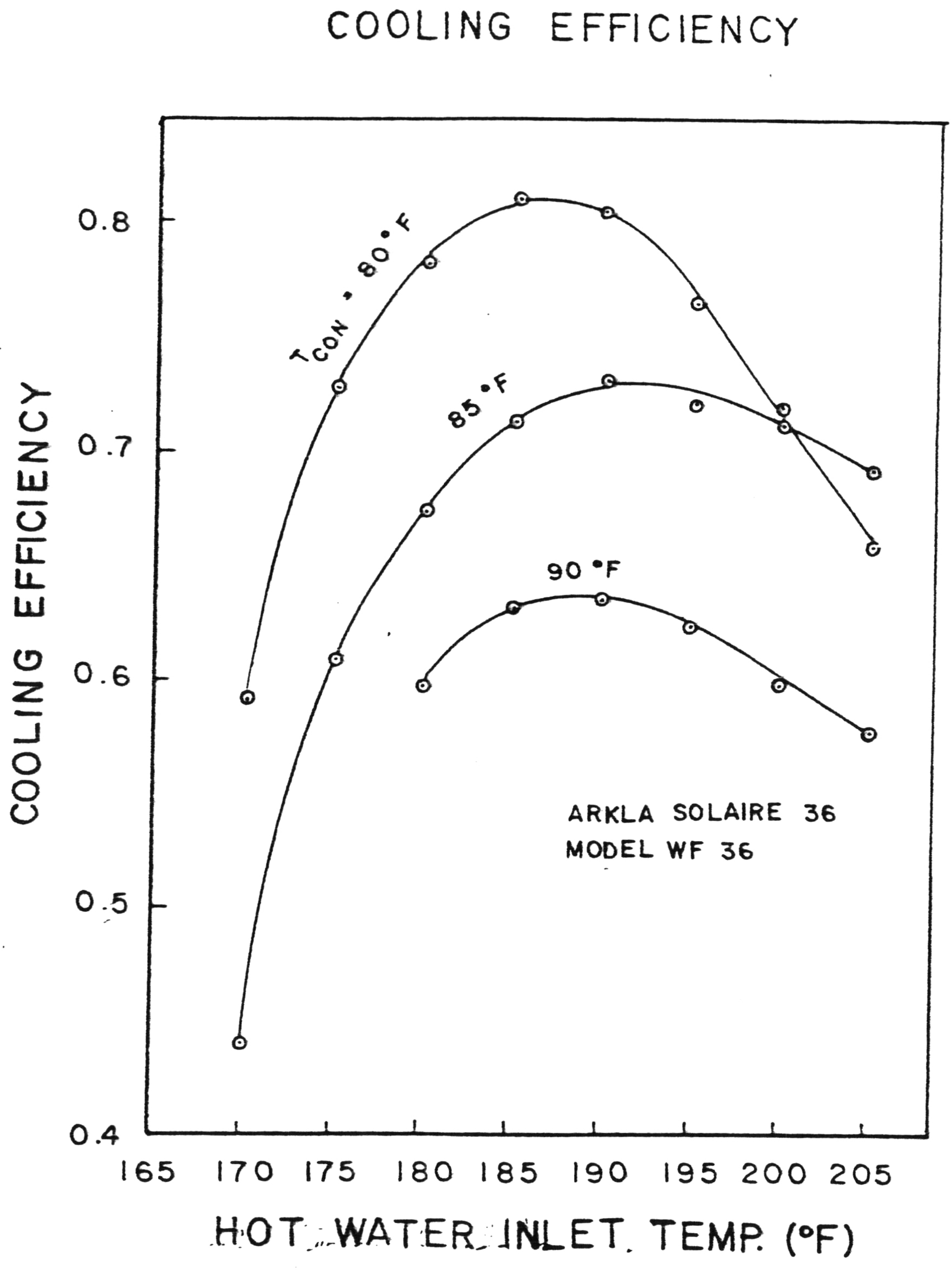


FIGURE B.7.4 EFFICIENCY OF ARKLA WF 36 THREE-TON UNIT

$$T_{con} = \begin{cases} 10 & \text{if } T_{wb} + 5.5 < 10^{o}C \\ 40 & \text{if } T_{wb} + 5.5 > 40^{o}C \\ T_{wb} + 5.5 & \text{elsewhere} \end{cases}$$

$$\dot{Q}_{cool} = \dot{Q}_{cap} \cdot f_1(T_i, T_{con})$$

$f_1(T_i, T_{con})$ = normalized cooling capacity, as shown in FIGURE B.7.2 ($f_1 \geq 0$)

$$f_1(T_i, T_{con}) = 0 \quad \text{if } T_i < 77^{o}C$$

$$\dot{Q}_{input} = \dot{Q}_{ref} \cdot f_2(T_i, T_{con})$$

$f_2(T_i, T_{con})$ = normalized input energy requirement rate as shown in FIGURE B.7.3 ($f_2 \geq 0.1$ if $\gamma = 1$)

$$f_2(T_i, T_{con}) = 0.1 \quad \text{if } T_i < 77^{o}C$$

$$\dot{Q}_{reject} = \dot{Q}_{cool} + \dot{Q}_{input}$$

$$T_o = T_i - \frac{\dot{Q}_{input}}{\dot{M}_i C_p}$$

<u>Multi-Stage Room Thermostat</u>

Let $T_{h2} > T_{h1} > T_{ho} > T_{co} > T_{c1} > T_{c2}$ be thermostat reference temperatures.

Let

T_i : inlet hot water temperature

T_{rm} : room temperature

T_{min} : minimum temperature of the hot water below which the absorption air-conditioner must not be operated

γ_{Ao}, γ_{Bo}, γ_{Co}

: previous values of control functions γ_A, γ_B, γ_C, respectively (γ = 0 or 1)

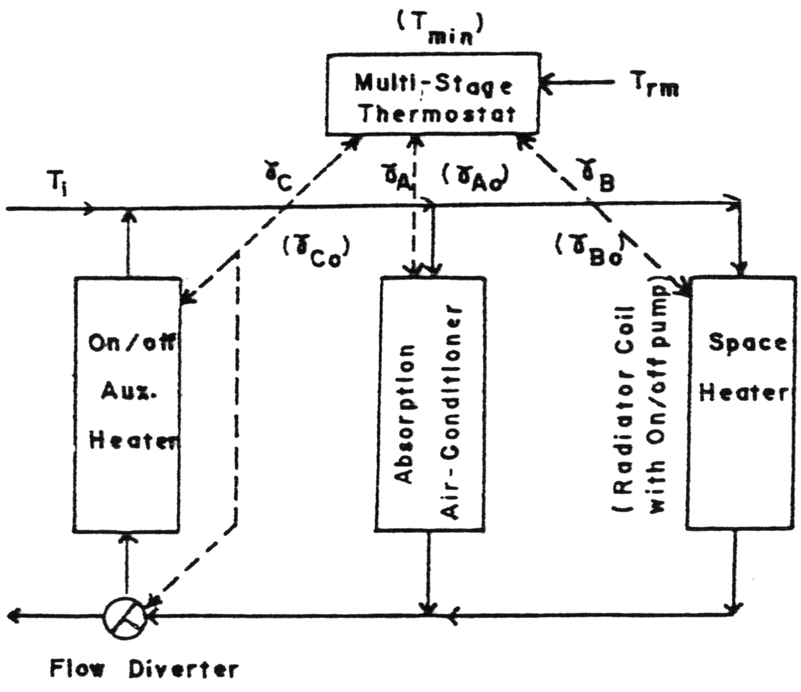


FIGURE B-8-1 ONE COMMON COMBINATION OF AN AUXILIARY HEATER, AN ABSORPTION AIR-CONDITIONER AND A SPACE HEATER

In view of numerical stability, the room temperature T_{rm} should not vary too much, relative to the width of each control band in TABLE B.8.1, during each integration step.

It should be noted that for the set up shown in FIGURE B.8.1 γ_C is also used to control the operation of the flow diverter so that no hot water will be drawn from or returned to the storage tank as long as the auxiliary heater is turned on. Eventhough the thermostat has been designed for ease of use for the setup in FIGURE B.8.1, a thorough understanding of its control mechanism allows it to be applied to other types of setups.

TABLE B.8.1 Values Taken by Thermostat Control Functions Under Various Circumstances

<table>
<tr><th>T_{rm}</th><th>γ_A</th><th>γ_B</th><th>γ_C</th></tr>
<tr><td>$T_{rm} > T_{h2}$</td><td>1</td><td>0</td><td>1</td></tr>
<tr><td rowspan="2">$T_{h1} \leq T_{rm} \leq T_{h2}$</td><td rowspan="2">1</td><td rowspan="2">0</td><td>1 if γ_{Co} = 1 or $T_i < T_{min}$</td></tr>
<tr><td>0 otherwise</td></tr>
<tr><td rowspan="2">$T_{ho} \leq T_{rm} < T_{h1}$</td><td>1 if γ_{Ao} = 1</td><td rowspan="2">0</td><td>0 if γ_{Ao} = 0 or γ_{Co} = 0</td></tr>
<tr><td>0 if γ_{Ao} = 0</td><td>1 otherwise</td></tr>
<tr><td>$T_{Co} < T_{rm} < T_{ho}$</td><td>0</td><td>0</td><td>0</td></tr>
<tr><td rowspan="2">$T_{C1} < T_{rm} \leq T_{Co}$</td><td rowspan="2">0</td><td>1 if γ_{Co} = 1</td><td>0 if γ_{Bo} = 0 or γ_{Co} = 0</td></tr>
<tr><td>0 if γ_{Co} = 0</td><td>1 otherwise</td></tr>
<tr><td rowspan="2">$T_{C2} \leq T_{rm} \leq T_{C1}$</td><td rowspan="2">0</td><td rowspan="2">1</td><td>1 if γ_{Co} = 1</td></tr>
<tr><td>0 if γ_{Co} = 0</td></tr>
<tr><td>$T_{rm} < T_{C2}$</td><td>0</td><td>1</td><td>1</td></tr>
</table>

Solar Radiation Processor

Several relationships are available for estimating the fraction of diffuse radiation from the total solar radiation on a horizontal surface. They are:

1. Liu and Jordan's Relationship:

This relationship was based on their study of daily diffuse and daily total solar radiation.[28]

Let

H_d	:	diffuse radiation flux on a horizontal surface
H	:	measured total radiation flux on a horizontal surface
K_t	:	H/H_o $(0 \leq K \leq 0.75)$
H_o	:	extraterrestrial total radiation flux on a horizontal surface $\left[= \left\{ 1 + 0.033 \cos(360n/365) \right\} S_c \cos(\theta_h) \right]$
n	:	day of the year
S_c	:	solar constant
θ_h	:	angle of incidence of beam radiation on the tilted surface (^{o})

$$\frac{H_d}{H} = 1.0045 + 2.6313K_t^3 - 3.5227K_t^2 + 0.04349K_t$$

Then

$$H_b = H - H_d$$

where

H_b = beam radiation flux on a horizontal surface

2. Boes' Relationship:

The following equations are used to obtain beam and diffuse radiation fluxes on a horizontal surface.(29)

$$H_{dn} = H_{direct\ normal} = 1800K_t - 520 \quad (\text{Watts/m}^2)$$

If $H_{dn} > 1000$, $H_{dn} = 1000$ (Watts/m^2)

If $H_{dn} < 0$, $H_{dn} = 0$ (Watts/m^2)

where

$$K_t = H/H_o \quad (0 \leq K_t \leq 1)$$

Then

$$H_b = H_{dn} \cos(\theta_h)$$

$$H_d = H - H_b$$

3. Orgill's Relationship:

The following correlation has been obtained by J.F. Orgill of University of Waterloo, Ontario, Canada, fromm his analysis of hourly diffuse radiation on a horizontal surface.(16)

$$K = \begin{cases} 1.0 - 0.249K_t & \text{if } 0 \leq K_t \leq 0.35 \\ 1.557 - 1.84K_t & \text{if } 0.35 \leq K_t \leq 0.75 \\ 0.177 & \text{if } K_t > 0.75 \end{cases}$$

$$H_d = KH$$

$$H_b = H - H_d$$

In this module, an option is provided to choose one of the above three relationships: option 1 for Liu and Jordan's relationship, 2 for Boes', and 3 for Orgill's. Option 4 is used when the diffuse and beam radiation fluxes on a horizontal surface are already known.

Let

R_b : the ratio of beam radiation on a tilted surface to that on a horizontal surface

H_B : beam radiation flux on the tilted surface

H_b : beam radiation flux on the horizontal surface

$$R_b = \cos(\theta_t)/\cos(\theta_h) = \frac{H_B}{H_b}$$

$$\cos(\theta_h) = \cos(\phi)\cos(\delta)\cos(\omega) + \sin(\phi)\sin(\delta)$$

$$\cos(\theta_t) = \cos(S)\cos(\delta)\sin(\phi) - \sin(\delta)\cos(\phi)\sin(S)\cos(\gamma) + \cos(\delta)\cos(\phi)\cos(S)\cos(\omega) + \cos(\omega)\cos(\delta)\sin(S)\cos(\gamma)\sin(\phi) + \cos(\delta)\sin(S)\sin(\gamma)\sin(\omega)$$

δ = solar declination ($^\circ$)

$= 23.45 \sin\{(284 + n)\, 360/365\}$

ω = solar hour angle ($^\circ$)

= 15(12 noon - solar hour of the day)

ϕ = latitude ($^\circ$), positive in the Northern hemisphere

S = angle between the tilted surface and the horizontal surface ($^\circ$) ($0 \leq S \leq 90$)

γ = orientation of the horizontal projection of the normal of the tilted surface, measured from the south and negative when clockwise ($^\circ$)

Let

R_d : the ratio of diffuse radiation flux on a tilted surface to that on a horizontal surface

H_D : diffuse radiation flux on a tilted surface (excluding ground reflection)

H_d : diffuse radiation flux on a horizontal surface

$$R_d = \frac{H_D}{H_d}$$

Let

R_r : the ratio of reflected diffuse radiation on a tilted surface to the total radiation on a horizontal surface

ρ : ground reflectance

$$R_r = \rho(1 - \cos(S))/2$$

In short,

$$H_B = R_b H_b$$

$$H_{DT} = H_d R_d + H R_r$$

$$H_T = H_B + H_{DT}$$

where

H_{DT} = total diffuse radiation flux on a tilted surface

H_T = total radiation flux on a tilted surface

Following the development of Duffie and Beckman,(19)

solar time (hours) = standard time + E + $(L_{st} + L_{loc})/15 + \Delta_s$

Here standard time is the local time at which radiation data are recorded. E is a correction factor (from the equation of time in hours) that accounts for the eccentricity of the earth's orbit and varies between approximately -0.24 hours and +0.26 hours each year. L_{st} is the standard meridian for the local time zone and L_{loc} is the longtitude of the location of interest (in degrees west). Δ_s is a third small correction factor that accounts for the time shift in the solar radiation data relative to the nominal value of the time of the reading.

If the reading is the integrated radiation preceding the reporting time, Δ_s is -0.5 hour.

According to Siebers,[30]

$$E = -[0.1236\sin(x) - 0.004289\cos(x) + 0.1539\sin(2x) + 0.06078\cos(2x)] \quad \text{(hours)}$$

where

$$x = 2\pi(n - 1)/365.242 \quad \text{(radians)}$$

$$n = \text{day of the year} \quad (\text{e.g. } n = 1 \text{ for January 1})$$

Space Heating and Cooling of a One-Node House

Let

$\dot{Q}_{load}$: rate of heat transfer from the surroundings into a house

UA : product of the house overall heat transfer coefficient and total heat transfer area

T_{amb} : ambient temperature

T_{rm} : average house temperature

$$\dot{Q}_{load} = UA(T_{amb} - T_{rm})$$

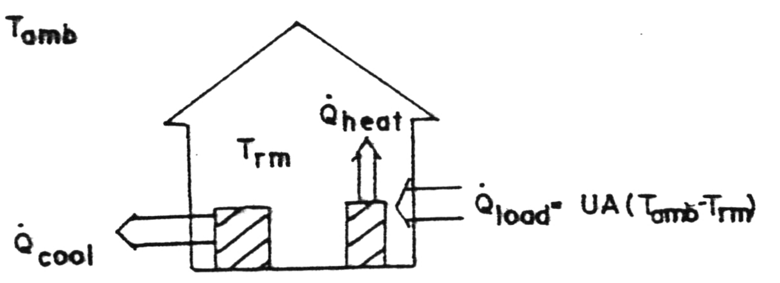


FIGURE B.9.1 SPACE HEATING AND COOLING OF A ONE-NODE HOUSE

Let

C : total heat capacity of the house

t : time

$\dot{Q}_{heat}$: rate of heat input into the house (Set $\dot{Q}_{heat}$ to be zero if no heat input device is used.)

$\dot{Q}_{cool}$: rate of heat removal from the house by a space cooling device (Set $\dot{Q}_{cool}$ to be zero if no such device is used.)

Then the energy balance for the house becomes

$$C \frac{dT_{rm}}{dt} = \dot{Q}_{load} + \dot{Q}_{heat} - \dot{Q}_{cool}$$

or

$$\frac{dT_{rm}}{dt} = (\dot{Q}_{load} + \dot{Q}_{heat} - \dot{Q}_{cool})/C$$

Flat-Plate Solar Collector (Unsteady State Model)

FIGURE B.10.1 shows a typical configuration of the flat-plate solar collector. Generally, the temperature T will be a function of time and space.

$$T = T(t,x,y,z)$$

The complexity of a resulting unsteady-state model can, however, be greatly reduced by a few preliminary considerations, as follows:

Let

T_f : fluid temperature

T_p : absorber plate temperature

T_g : glass cover temperature

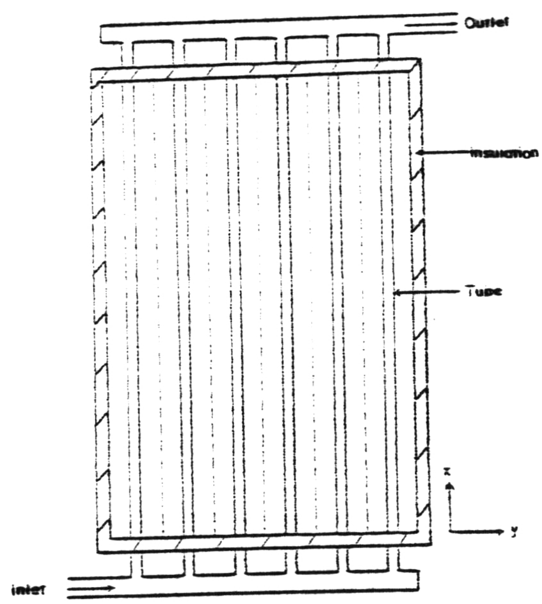
T_d : bottom insulation temperature

We have

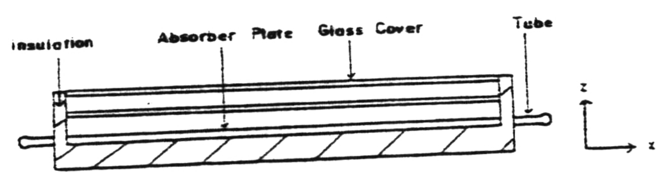
$T_f = T_f(t,x)$ (forced fluid circulation in tubes & plug-flow)

$T_p = T_p(t,x)$ (a small thickness, good thermal conductor, width of a segment is small compared to its length, temperature gradient is in the flow direction, say, x-direction)

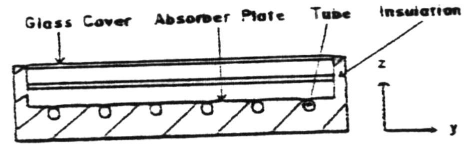
$T_g = T_g(t)$ (all of the main heat transfer modes



(a) TOP VIEW OF A FLAT-PLATE SOLAR COLLECTOR



(b) SIDE VIEW OF A FLAT-PLATE SOLAR COLLECTOR



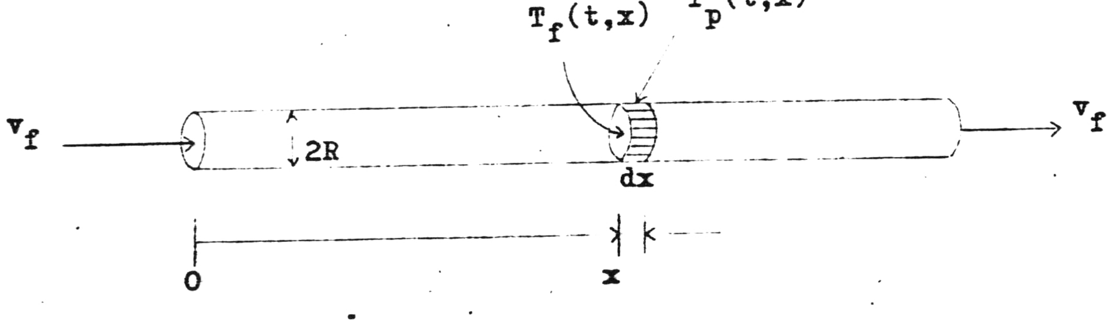
(c) FRONT VIEW OF A FLAT-PLATE SOLAR COLLECTOR

FIGURE 3.10-1 A TYPICAL CONFIGURATION OF THE FLAT-PLATE SOLAR COLLECTOR
(a) TOP VIEW, (b) SIDE VIEW,
(c) FRONT VIEW

are not directionally specific, the thermal performance of the collector is affected very little by the temperature distribution in its glass covers)

$T_d = T_d(t)$ (the temperature distribution in the bottom insulation have no direct effect on the thermal performance of the collector, the rate of heat loss through the insulation and the rate of heat absorption by the insulation itself can be eliminated, i.e., the insulation is treated as a lumped-parameter system characterized by an average temperature T_d)

1. Tube Energy Balance:



(rate of heat accumulation) = (rate of heat in by conduction)
-(rate of heat out by conduction)
+(rate of heat transfer through

tube wall)

+(rate of heat in by convection)

-(rate of heat out by convection)

The energy balance on a tube section of thickness dx in t the above figure and rearranged is given by

$$\frac{\partial T_f}{\partial t} = \frac{1}{\rho_f c_f}\left[k_f \frac{\partial^2 T_f}{\partial x^2} + \frac{2U_f}{R}(T_p - T_f) - \rho_f c_f v_f \frac{\partial T_f}{\partial x}\right]$$

where

ρ_f	:	density of fluid
c_f	:	specific heat of fluid
R	:	radius of tube
k_f	:	thermal conductivity of fluid
U_f	:	overall heat transfer coefficient between absorber plate and tube fluid
v_f	:	fluid velocity in the tube

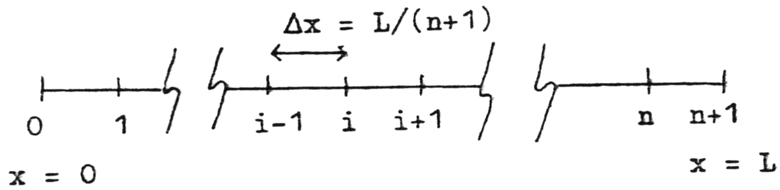
By means of the finite difference method, we replace

$$\frac{\partial T_f}{\partial x} \quad \text{by} \quad \frac{T_{f_{i+1}} - T_{f_{i-1}}}{2\Delta x}$$

and

$$\frac{\partial T_f}{\partial x^2} \quad \text{by} \quad \frac{1}{\Delta x^2}(T_{f_{i+1}} - 2T_{f_i} + T_{f_{i-1}})$$

where T_{f_i} is the value of T_f at mesh point i.



Then the above equation becomes, for $1 \leq i \leq n$,

$$\frac{dT_{f_i}}{dt} = \frac{1}{\rho_f c_f}\left[\frac{k_f}{\Delta x^2}\left(T_{f_{i+1}} - 2T_f + T_{f_{i-1}}\right) + \frac{2U_f}{R}\left(T_{p_i} - T_{f_i}\right) - \frac{\rho_f c_f v_f}{2\Delta x}\left(T_{f_{i+1}} - T_{f_{i-1}}\right)\right]$$

This n ordinary differential equations can be solved simultaneously on a digital computer via a suitable algorithm.

The initial condition and boundary conditions in terms of finite-difference scheme are:

$$T_{f_i} = T_f(0,i) = f(i) \qquad \text{for } 0 \leq i \leq n+1 \quad \text{at } t = 0$$

$$T_{f_o} = T_f(t,0) = T_{f_{inlet}} \qquad \text{for } t > 0$$

$$T_{f_{n+1}} = T_f(t,n+1) = T_{f_{outlet}} \qquad \text{for } t > 0$$

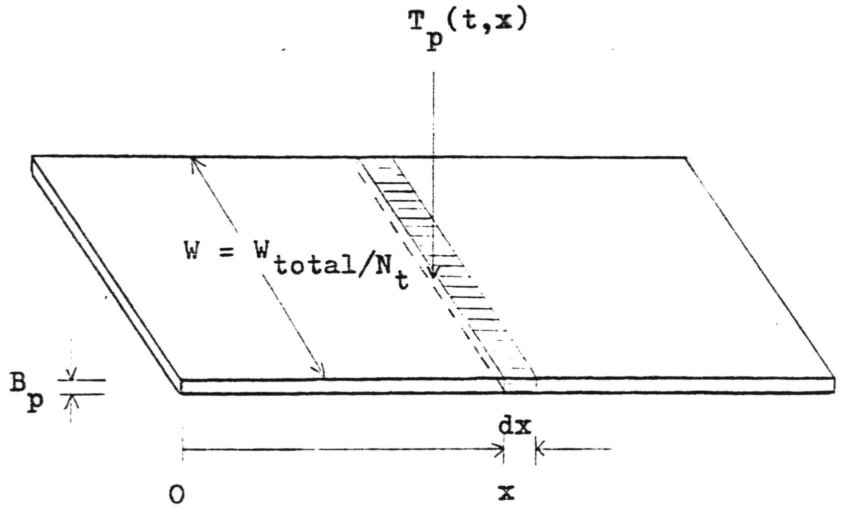
Assuming the temperature profile is flat over the cross section at the outlet, we have

$$T_{f_{outlet}} = T_{f_{n+1}} = T_{f_n}$$

Then, the set of n ordinary differential equations can be solved simultaneously.

2. Absorber Plate Energy Balance:

As previously mentioned, it is only necessary to consider just one symmetrical segment of the absorber plate. There are as many such plates as there are tubes, the total number of which is N_t.



For a section of thickness dx of the absorber plate,

$$\begin{pmatrix}\text{rate of heat}\\ \text{accumulation}\end{pmatrix} = \begin{pmatrix}\text{rate of heat in}\\ \text{by conduction}\end{pmatrix} - \begin{pmatrix}\text{rate of heat out}\\ \text{by conduction}\end{pmatrix}$$

$$+\begin{pmatrix}\text{rate of solar}\\ \text{energy absorption}\end{pmatrix} - \begin{pmatrix}\text{rate of heat}\\ \text{reradiation}\end{pmatrix}$$

$$-\begin{pmatrix}\text{rate of heat}\\ \text{convection to}\\ \text{glass cover}\end{pmatrix} - \begin{pmatrix}\text{rate of heat}\\ \text{loss through}\\ \text{bottom insulation}\end{pmatrix}$$

$$- \begin{pmatrix} \text{rate of heat transfer} \\ \text{to tube fluid} \end{pmatrix}$$

Mathematically and rearrange,

$$\frac{\partial T_p}{\partial t} = \frac{1}{\rho_p c_p B_p} \left[B_p k_p \frac{\partial^2 T_p}{\partial x^2} + \lambda_p Q_s - \sigma \epsilon_{pg_1} (T^4_{p_{abs}} - T^4_{g_{1,abs}}) - h_{pg_1}(T_p - T_{g_1}) - U_d(T_p - T_d) - \frac{2\pi R}{W} U_f(T_p - T_f) \right]$$

where the subscript "p" refers to the absorber plate and the subscript "g_1" refers to the glass cover adjacent to the absorber plate.

Here

B_p : thickness of absorber plate

$\lambda_p^{(31)}$: fraction of total solar radiation absorbed by the absorber plate

Q_s : total solar radiation flux incidental on the collector surface

σ : Stefan-Boltzmann constant

ϵ_{pg_1} : effective emissivity for radiation heat exchange between the absorber plate and the adjacent glass cover

$T_{p_{abs}}$: absolute temperature of absorber plate in degrees Kelvin or Rankine

$T_{g_{1,abs}}$: absolute temperature of adjacent glass cover

$h_{pg_1}^{(13)}$: convective heat transfer coefficient between the absorber plate and the adjacent glass cover

U_d : overall heat transfer coefficient between the absorber plate and the bottom insulation

$U_f^{(25)}$: overall heat transfer coefficient between the absorber plate and the tube fluid

W : width of each symmetrical absorber plate segment

The effective emissivity, ϵ_{ij}, for radiative heat exchange between surfaces i and j is given by

$$\epsilon_{ij} = \frac{1}{(1/\epsilon_i + 1/\epsilon_j - 1)}$$

where

ϵ_i, ϵ_j : emissivity of surfaces i and j, respectively.

Using the same finite-difference scheme as in the tube energy balance, the above partial differential equation can be transformed as follows:

For $1 \leq i \leq n$,

$$\frac{dT_{p_i}}{dt} = \frac{1}{\rho_p c_p B_p} \Big[\frac{k_p B_p}{\Delta x^2}(T_{p_{i+1}} - 2T_{p_i} + T_{p_{i-1}}) + \lambda_p Q_s$$

$$- \sigma \epsilon_{pg_1}(T^4_{p_{i,abs}} - T^4_{g_{1,abs}})$$

$$- h_{pg_1}(T_{p_i} - T_{g_1}) - U_d(T_{p_i} - T_d)$$

$$- \frac{2\pi R}{W} U_f (T_{p_i} - T_{f_i}) \Big]$$

with the initial condition and boundary conditions,

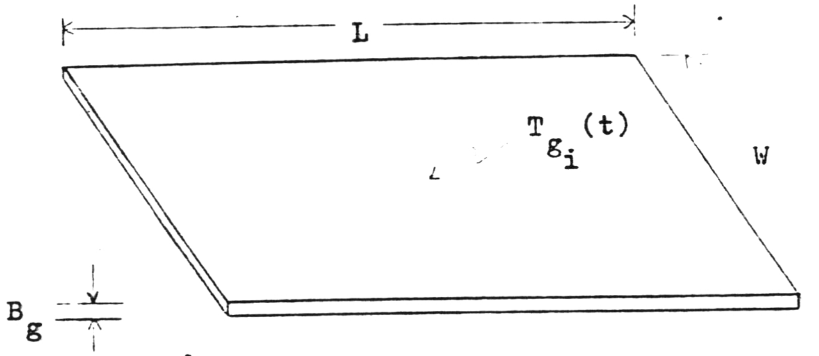
$$T_{p_i} = f_2(i) \quad \text{at } t = 0$$

$$T_{p_{n+1}} = T_{p_n} \quad \text{for } t > 0$$

$$T_{p_o} = T_{p_1} \quad \text{for } t > 0$$

3. Glass Covers Energy Balance:

An option is available for selecting the number of glass covers, N_G ($1 \le N_G \le 4$). The higher the number of glass covers, the higher the collector fluid temperature. For simplicity, only one segment of the glass cover corresponding to the absorber plate segment will be considered. Also all glass covers are assumed to be identical in shape and properties.



3.1 Only One Glass Cover ($N_G = 1$),

$$\begin{pmatrix}\text{rate of heat}\\ \text{accumulation}\end{pmatrix} = \begin{pmatrix}\text{rate of solar}\\ \text{radiation}\\ \text{absorbed}\end{pmatrix} + \begin{pmatrix}\text{rate of reradiation}\\ \text{from the absorber}\\ \text{plate}\end{pmatrix}$$

$$+\begin{pmatrix}\text{rate of convective}\\ \text{heat transfer from}\\ \text{the absorber plate}\end{pmatrix} - \begin{pmatrix}\text{rate of reradiation}\\ \text{from the glass cover}\\ \text{to the sky}\end{pmatrix}$$

$$-\begin{pmatrix}\text{rate of convective}\\ \text{heat loss to the}\\ \text{surroundings}\end{pmatrix}$$

Let

λ_{g_i} : fraction of total incoming solar radiation flux absorbed by glass cover i

$T_{g_{i,abs}}$: absolute temperature of glass cover i

$\epsilon_{g_{is}}$: effective emissivity for radiative heat exchange between glass cover i and the sky

$h_w^{(25)}$: convective heat transfer coefficient of the wind blowing past the top glass cover

$T_{s,abs}$: absolute temperature of the sky

T_{amb} : ambient temperature

Δx : mesh size ($\Delta x = L/(n + 1)$)

subscript "g" refers to any glass cover

subscript "g_i" refers to the i-th glas cover (counting upwards)

subscript "s" refers to the sky

The summation sign $\sum_i$ is defined here as

$$\sum_i f_i = \tfrac{1}{2}(f_o + f_{n+1}) + \sum_{i=1}^{n} f_i$$

Mathematically and rearrange,

$$\frac{dT_{g_1}}{dt} = \frac{1}{\rho_g c_g B_g}\Big[\lambda_{g_1} Q_s + \frac{\sigma \epsilon_{pg_1}}{(n+1)} \sum_i (T^4_{p_{i,abs}} - T^4_{g_{1,abs}})$$
$$+ \frac{h_{pg_1}}{(n+1)} \sum_i (T_{p_i} - T_{g_1})$$
$$- \sigma \epsilon_{g_1 s}(T^4_{g_{1,abs}} - T^4_{s,abs})$$
$$- h_w(T_{g_1} - T_{amb})\Big]$$

3.2 Two or More Glass Covers ($N_G \geq 2$),

For the bottom glass cover (i = 1),

(rate of heat accumulation) = (rate of solar radiation absorbed by the glass cover) + (rate of reradiation from the absorber plate to the glass cover)

+ (rate of convective heat transfer from the absorber plate) - (rate of reradiation from this glass cover to an adjacent cover)

- (rate of convective heat transfer to an adjacent cover)

Mathematically and rearrange,

$$\frac{dT_{g_1}}{dt} = \frac{1}{\rho_g c_g B_g}\Bigg[\lambda_{g_1} Q_s + \frac{\sigma\epsilon_{pg_1}}{(n+1)} \sum_i (T^4_{p_i,abs} - T^4_{g_1,abs})$$

$$+ \frac{h_{pg_1}}{(n+1)} \sum_i (T_{p_i} - T_{g_1}) - \sigma\epsilon_{g_1 g_2}(T^4_{g_1,abs} - T^4_{g_2,abs})$$

$$- h_{g_1 g_2}(T_{g_1} - T_{g_2})\Bigg]$$

Similarly for the top glass cover ($i = N_G$),

$$\frac{dT_i}{dt} = \frac{1}{\rho_g c_g B_g}\Bigg[\lambda_i Q_s + \sigma\epsilon_{g_{i-1} g_i}(T^4_{g_{i-1},abs} - T^4_{g_i,abs})$$

$$+ h_{g_{i-1} g_i}(T_{g_{i-1}} - T_{g_i})$$

$$- \sigma\epsilon_{g_i s}(T^4_{g_i,abs} - T^4_{s,abs})$$

$$- h_w(T_{g_i} - T_{amb})\Bigg]$$

For any glass cover, if one exists, between the top and bottom cover ($2 \le i \le N_G - 1$);

$$\frac{dT_{g_i}}{dt} = \frac{1}{\rho_g c_g B_g}\Bigg[\lambda_i Q_s + \sigma\epsilon_{g_{i-1} g_i}(T^4_{g_{i-1},abs} - T^4_{g_i,abs})$$

$$+ h_{g_{i-1}g_i}(T_{g_{i-1}} - T_{g_i})$$

$$+ \sigma\epsilon_{g_i g_{i+1}}(T^4_{g_{i,abs}} - T^4_{g_{i+1,abs}})$$

$$- h_{g_i g_{i+1}}(T_{g_i} - T_{g_{i+1}})\Big]$$

4. Bottom Insulation Energy Balance:

For simplicity, the effects of the edges and corners are assumed insignificant with respect to the estimation of overall rate of heat loss through the bottom insulation.

Let

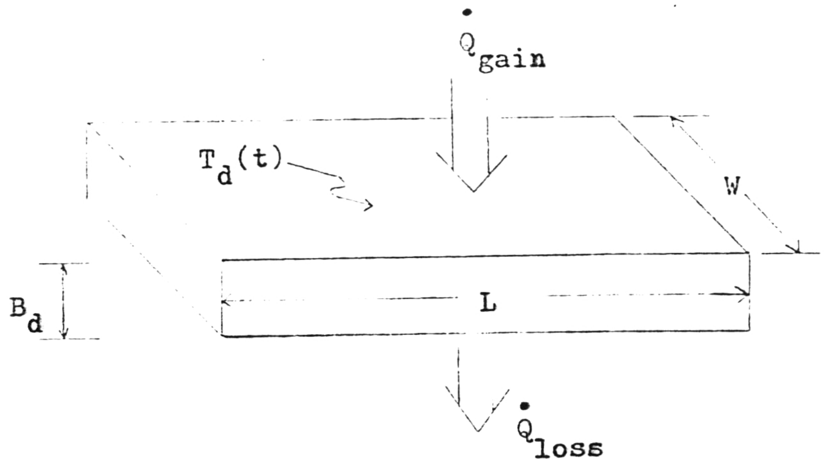
$T_{p,avg}$: average temperature of the absorber plate

U_{d_1} : overall heat transfer coefficient between the absorber plate and the insulation

U_{d_2} : overall heat transfer coefficient between the insulation and the surroundings below it

T_{amb} : ambient temperature of the surroundings below the insulation

the subscript "d" refers to the insulation.



$$\begin{pmatrix}\text{rate of heat}\\ \text{accumulation}\end{pmatrix} = \begin{pmatrix}\text{rate of heat}\\ \text{gain}\end{pmatrix} - \begin{pmatrix}\text{rate of heat}\\ \text{loss}\end{pmatrix}$$

Mathematically,

$$\frac{dT_d}{dt} = \frac{1}{\rho_g c_d B_d}\left[U_{d_1}(T_{p,avg} - T_d) - U_{d_2}(T_d - T_{amb})\right]$$

For a well-designed solar collector the most dominant resistance to heat loss should always be the insulation because it is put there primarily to hinder heat loss. Therefore, a good approximation is

$U_{d_1} \simeq k_d/B_{d,eff}$

and

$U_{d_2} \simeq k_d/B_{d,eff}$

where

k_d : thermal conductivity of the insulation

$B_{d,eff}$: effective thickness of the insulation

It is reasonable to assume that the average insulation temperature represents a temperature somewhere near the middle thickness of the insulation. In this case,

$$B_{d,eff} = \tfrac{1}{2}B_d$$

Summary:

The unsteady-state solar collector is modelled by a set of (2(n - 1) + N_G + 1) ordinary differential equations, where n is the number of mesh points excluding end points, and N_G is the number of glass cover (between 1 and 4). A value of n arround 10 is recommended for use in simulation.

Heat Exchanger

The following mathematical description of a zero-capacitance sensible heat exchanger is discussed in detail by Kays and London.[11] There are five optional modes of operation: 1. parallel flow, 2. counter flow, 3. cross flow with hot-side fluid unmixed but cold-side completely mixed, 4. cross flow with hot-side fluid completely mixed but cold-side unmixed, and 5. constant heat transfer effectiveness. For the first four modes, a constant overall heat transfer coefficient is given together with the hot- and cold-side inlet temperatures and flow rates. From these the heat transfer effectiveness E is calculated, depending on the particular mode of operation. Then, the hot- and cold-side outlet temperatures and the rate of heat transfer across the exchanger are computed as follows:

$$T_{h_o} = T_{h_i} - E\left(\frac{C_{min}}{C_h}\right)(T_{h_i} - T_{c_i})$$

$$T_{c_o} = E\left(\frac{C_{min}}{C_c}\right)(T_{h_i} - T_{c_i}) + T_{c_i}$$

$$\dot{Q}_T = EC_{min}(T_{h_i} - T_{c_i})$$

where

T_{h_i}, T_{h_o} : hot-side inlet and outlet fluid temperatures, respectively

T_{c_i}, T_{c_o} : cold-side inlet and outlet fluid temperatures, respectively

E : heat transfer effectiveness

$\dot{Q}_T$: rate of heat transfer across the heat exchanger

C_c : capacity rate of cold-side fluid

C_h : capacity rate of hot-side fluid

C_{min} : minimum capacity rate

$\dot{M}_c$: mass flow rate of cold-side fluid

$\dot{M}_h$: mass flow rate of hot-side fluid

c_{p_c} : specific heat of cold-side fluid

c_{p_h} : specific heat of hot-side fluid

$$C_{min} = \min\{C_c, C_h\}$$

where

$$C_c = \dot{M}_c c_{p_c}$$

$$C_h = \dot{M}_h c_{p_h}$$

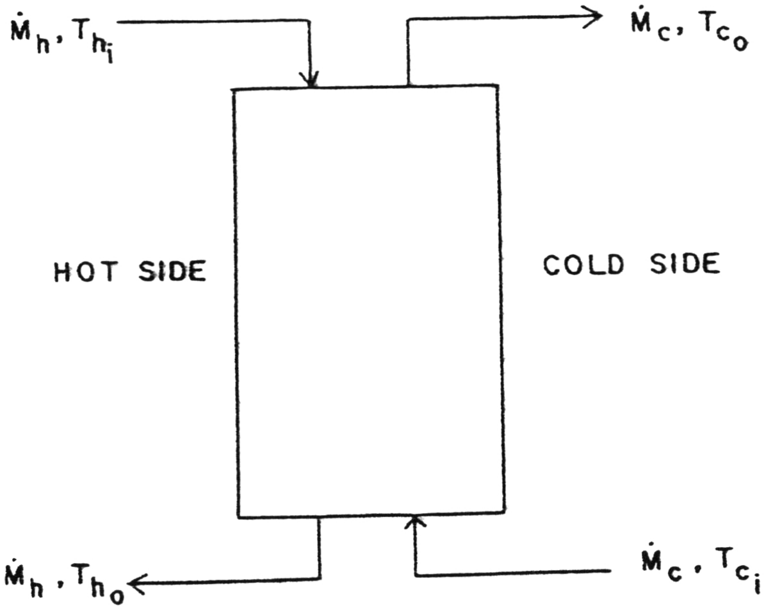


FIGURE B.II.1 HEAT EXCHANGER(COUNTER FLOW)

Mode 1 : Parallel Flow

$$E = \frac{1 - e^{-UA\left(1 + \frac{C_{min}}{C_{max}}\right)/C_{min}}}{1 + \frac{C_{min}}{C_{max}}}$$

where

$$C_{max} = \max\left\{C_c, C_h\right\}$$

Mode 2 : Counter Flow

$$E = \frac{1 - \exp\left[- UA\left(1 - \frac{C_{min}}{C_{max}}\right)/C_{min}\right]}{1 - \left(\frac{C_{min}}{C_{max}}\right)\exp\left[- UA\left(1 - \frac{C_{min}}{C_{max}}\right)/C_{min}\right]}$$

Mode 3 : Cross Flow (Hot-side Unmixed, Cold-side Mixed)

If $C_{max} = C_h$,

$$E = 1 - e^{- rC_{max}/C_{min}}$$

where

$$r = 1 - e^{- UA/C_{max}}$$

If $C_{min} = C_h$,

$$E = (C_{max}/C_{min})\,(1 - e^{- rC_{min}/C_{max}})$$

where

$$r = 1 - e^{-UA/C_{min}}$$

Mode 4 : Cross Flow (Hot-side Mixed, Cold-side Unmixed)

If $C_{max} = C_c$,

$$E = 1 - e^{- rC_{max}/C_{min}}$$

where

$$r = 1 - e^{-UA/C_{max}}$$

If $C_{min} = C_c$,

$$E = (C_{max}/C_{min})\ (1 - e^{-rC_{min}/C_{max}})$$

where

$$r = 1 - e^{-UA/C_{min}}$$

Mode 5 : Constant Effectiveness

In contrast to the first four modes, the effectiveness of the heat exchanger is simply specified and the concept of an overall heat transfer coefficient is not used.

If $C_{min} = C_h$,

$$\dot{Q}_{max} = C_h(T_{h_i} - T_{c_i})$$

If $C_{min} = C_c$,

$$\dot{Q}_{max} = C_c(T_{h_i} - T_{c_i})$$

Furthermore,

$$\dot{Q}_T = E\dot{Q}_{max}$$

$$T_{h_o} = T_{h_i} - \frac{\dot{Q}_T}{C_h}$$

$$T_{c_o} = T_{c_i} + \frac{\dot{Q}_T}{C_c}$$

Special Cases

Mode 2 :

If $\left|\frac{C_{min}}{C_{max}} - 1.0\right| < 0.01$,

$$E \simeq \frac{UA/C_{min}}{1 + \frac{UA}{C_{min}}}$$

Mode 1, 2, 3 and 4 :

If $\frac{C_{min}}{C_{max}} \leq 0.01$

$$E \simeq 1 - e^{-UA/C_{min}}$$

Data Reader

For speedy data reading, all data sets must conform to a single format. The applicable format is specified by an array named FMT. No provision is available for reading in a "free" format. Furthermore, only one data reader may be used, but the data set may contain as many numbers as desired.

On/off Differential Controller with Hysteresis

This controller generates a control variable γ_o with a value of either 0 or 1. The value of γ_o depends on the temperature difference $\Delta T = T_1 - T_2$, two reference temperature T_A and T_B ($T_A < T_B$) and γ_i, the previous value of γ_o, as follows:

If $\gamma_i = 1$ and $\Delta T \geq T_A$,

$\gamma_o = 1$

If $\gamma_i = 1$ and $\Delta T < T_A$,

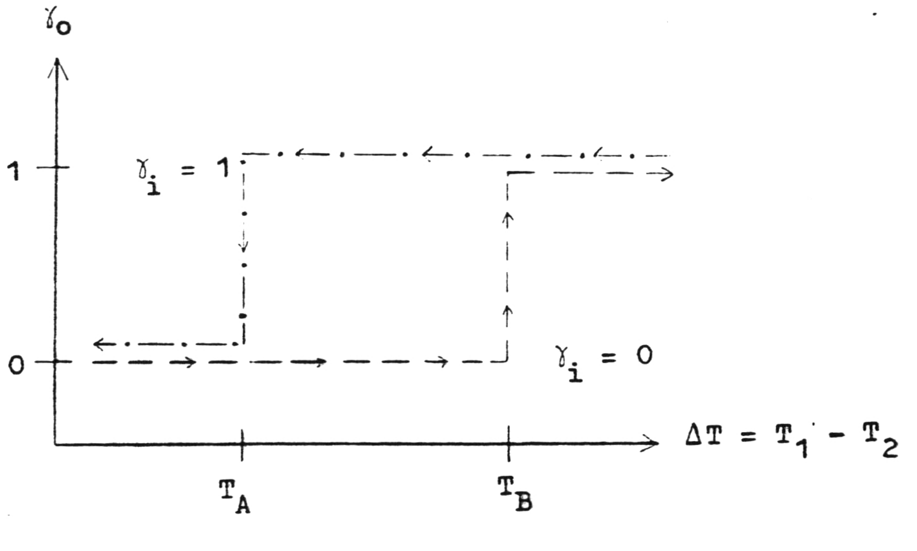
$\gamma_o = 0$

If $\gamma_i = 0$ and $\Delta T \geq T_B$,

$\gamma_o = 1$

If $\gamma_i = 0$ and $\Delta T < T_B$,

$\gamma_o = 0$



It should be noted that the controller is not restricted to sensing temperature and that hysteresis occurs only when γ_i is the previous value of γ_o. Furthermore, $\dot{T}_2$ can be set equal to 0 if a temperature, instead of a temperature difference is of interest.

Flat-Plate Solar Collector (Quasi-Steady-State Method)

A modified version of TRNSYS flat-plate collector model,[2] the model has instantaneous response to any input variation and is based on the work of Hottel, Whillier, and Bliss.[8],[9],[10]

The model expresses the rate of energy collection, $\dot{Q}_u$, as

$$\dot{Q}_u = AF_R\left[H_T(\tau\alpha) - U_L(T_i - T_a)\right] = \dot{m}C_p(T_o - T_i)$$

where

$$F_R = \frac{\dot{m}C_p}{AU_L}\left[1 - \exp\left(-\frac{F'U_LA}{\dot{m}C_p}\right)\right]$$

Here

A	:	area of solar collector
F_R	:	collector efficiency factor
H_T	:	total solar radiation flux striking the tilted collector surface
$\tau\alpha$[8]	:	fraction of the total solar flux absorbed by the collector plate
	:	product of the transmittance of the N glass covers

and the collector plate absorptance for solar radiation

U_L[32] : collector overall heat transfer coefficient

T_i : collector fluid inlet temperature

T_a : ambient temperature

$\dot{m}$: mass flow rate of collector fluid

C_p : heat capacity of collector fluid

F' : collector geometry factor (dimensionless)

F' is a function of the collector construction and can be determined in the manner given by Bliss[10] or Duffie and Beckman.[19]

When there is no fluid flow through the collector (when the pump is off), the collector fluid outlet temperature T_o is taken to be the average collector plate temperature as follows:

$$T_o = T_p = \frac{H_T(\tau\alpha)}{U_L} + T_a$$

where

T_p : mean collector plate temperature

<u>Mathematical Manipulator</u>

There are two modes of mathematical manipulation. In mode I, a single output is the result of a single mathematical operation on the first one or two inputs. Mode II employs the RPN (Reverse Polish Notation) to carry out a series of mathematical manipulations and generate a set of one or more outputs.

The Mode and the first mathematical manipulation is determined by the value of the first operational parameter, P_1.

Let

N_p : number of operational parameters, which is total number of parameters minus the first two

N_I : number of inputs values

P_j : value of the j-th operational parameter

S_k : value at the top of the operational stack

x_i : value of the i-th input

Y : output value and result of mathematical operations

Mode I : $P_1 \neq 0$ and $P_1 \neq -1$

If $P_1 = 1$, $Y = x_1 * x_2$

If $P_1 = 2$, $Y = x_1/x_2$

If $P_1 = 3$, $Y = x_1 + x_2$

If $P_1 = 4$, $Y = x_1 - x_2$

If $P_1 = 5$, $Y = x_1 ** x_2$

If $P_1 = 6$, $Y = \log_{10} x_1$

If $P_1 = 7$, $Y = - x_1$

If $P_1 = 8$, $Y = x_1$ if $x_1 \geq 0$
$= 0$ if $x_1 < 0$

If $P_1 = 9$, $Y = 1$ if $x_1 \geq x_2$
$= 0$ if $x_1 < x_2$

Boolean Functions (x_1, x_2, $Y = 0$ or 1)

If $P_1 = 10$, $Y = \bar{x}$ (NOT)

If $P_1 = 11$, $Y = x_1 \odot x_2$ (AND)

If $P_1 = 12$, $Y = x_1 \oplus x_2$ (OR)

Mode II : $P_1 = 0$ or $P_1 = -1$

Mode II uses an operational stack of up to 30 values to store the inputs as well as the results of previous mathematical manipulations. The operational parameters and inputs are used sequentially. As usual, the value of each operational parameter specifies what mathematical manipulation to carry out. All manipulations are performed on the stack top value if the manipulation is unary, and on the stack top 2 values if the manipulation is binary. After all operational parameters have been processed, the stack top value must become the last output, stored as the last element of the output array. If the index k of S_k is not equal to 1 or not all of the inputs have been used, an error message is printed to notify the user of improper mathematical manipulations. If division by zero is attempted, the simulation is terminated with an explanatory error message.

If $P_j = -1$, the value of the next parameter is placed on top of the stack as a constant ($j = j+1$, $S_k = P_j$).

If $P_j = 0$, the value of the next input is placed on top of the stack ($i = i+1$, $S_k = x_i$).

If $P_j = 1$, the stack top 2 values are replaced by their product ($S_{k-1} = S_{k-1} * S_k$).

If $P_j = 2$, the stack top 2 values are replaced by their division ($S_{k-1} = S_{k-1}/S_k$).

If $P_j = 3$, the stack top 2 values are replaced by their sum ($S_{k-1} = S_{k-1} + S_k$).

If $P_j = 4$, the stack top 2 values are replaced by their difference ($S_{k-1} = S_{k-1} - S_k$).

If $P_j = 5$, the stack top 2 values are replaced by $S_{k-1} = S_{k-1} ** S_k = (S_{k-1})^{S_k}$.

If $P_j = 6$, the stack top value is replaced by the logarithm ($S_k = \log_{10} S_k$).

If $P_j = 7$, the stack top value is negated ($S_k = - S_k$).

If $P_j = 8$, the stack top value is left unchanged, unless it is negative, in which case it is replaced by zero ($S_k = S_k$ if $S_k \geq 0$; $S_k = 0$ if $S_k < 0$).

If $P_j = 9$, the stack top 2 values are replaced by 1 if the second value is greater than the top value; otherwise, by zero ($S_{k-1} = 1$ if $S_{k-1} \geq S_k$; $S_k = 0$, otherwise).

Boolean Functions (S_k and/or $S_{k-1} = 0$ or 1)

If $P_j = 10$, the stack top value is replaced by its Boolean NOT ($S_k = \bar{S}_k$).

If $P_j = 11$, the stack top 2 values are replaced by their Boolean AND ($S_{k-1} = S_{k-1} \odot S_k$).

If $P_j = 12$, the stack top 2 values are replaced by their Boolean OR ($S_{k-1} = S_{k-1} \oplus S_k$).

Additional Mathematical Manipulations :

If $P_j = -2$, put the current simulation time on top of the stack

If $P_j = -3$, the stack top value is set as the next output

($j = j+1$, $Y_j = S_k$)

If $P_j = -4$, like $P_j = -3$, but the stack top value is removed

($j = j+1$, $Y_j = S_k$, $k = k-1$)

If $P_j = -5$, decrement the stack top pointer by 1 ($k = k-1$)

If $P_j = -6$, increment the stack top pointer by 1 ($k = k+1$)

If $P_j = -10 - i$,

place the value of the i-th input on top of the stack ($S_k = x_i$ where $i \geq 1$)

Wall(s) or Flat Roof

Sets of coefficients that define the thermal characteristics of nearly 100 different types of walls are listed in chapter 22 of ASHRAE Handbook of Fundamentals.[12] Because empirical equations have been built into this model, the specified units must be used.

Let

A_i : total area of wall or surface i, including window area (m^2)

b_n : transfer function coefficient based on a temperature difference term at time n (Btu/hr ft^2 ^oF)

d_n : transfer function coefficient of heat flux term at time n (-)

F_{is} : fraction of window area i that is shaded (-)

F_{iw} : fraction of wall i that is window area (-)

h_{ir} : radiation heat transfer coefficient (kJ/m^2 hr ^oC)

h_o : heat transfer coefficient by radiation and convection at the outer surface (kJ/m^2 hr ^oC)

h_w(25) : outside film heat transfer coefficient by convection (kJ/m^2 hr ^oC)

H_{T_i} : total incoming solar radiation flux incident on wall or surface i (kJ/m^2 hr)

N_B : number of b_n coefficients needed (-)

N_D : number of d_n coefficients needed (-)

N_G : number of glazings in each window (-)

$\dot{Q}_c$: conduction heat gains through walls and windows (kJ/hr)

$\dot{Q}_{SHG}$: rate of solar heat gain (kJ/hr)

$\dot{Q}_T$: total rate of heat gain (loss) to the room (kJ/hr)

q''_o : current heat flux entering (or leaving) the room (kJ/m^2 hr)

q''_n : heat flux entering (or leaving) the room at time n

$\bar{T}$: $\frac{1}{2}$ ($T_{amb} + T_{wall}$) (^oK)

t_{amb} : ambient temperature (^oC)

T_{sky}(25) : sky absolute temperature (^oK)

t_{rc} : room temperature (^oC)

$t_{sa,n}$: solar air temperature of a surface at time n (^oC)

W : wind speed (m/sec)

α : solar absorptance of an exterior surface ($0 \le \alpha \le 1$) (-)

ϵ_i : infrared emittance of surface i ($0 = \epsilon_i = 1$) (-)

σ : Stefan-Boltzmann constant ($\sigma = 2.041 \times 10^{-7}$ kJ/hr m^2 $°K^4$)

$\bar{\tau}$: effective transmittance of window ($0 = \bar{\tau} = 1$) (-)

The current heat flux into (or out of) a room, q''_o, can be computed as follows:

$$q''_o = \sum_{n=0} b_n (t_{sa,n} - t_{rc}) - \sum_{n=1} d_n q''_n \qquad (kJ/m^2\ hr)$$

The solar air temperature, t_{sa}, is defined as " that temperature of the outdoor air which, in the absence of all radiation exchanges, would give the same rate of heat entry into the surface as would exist with the actual combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air."

$$t_{sa} = t_{amb} + \frac{\alpha H_T}{h_o} + \frac{\epsilon \Delta R}{h_o} \qquad (°C)$$

where ΔR is " the difference between the longwave radiation incident on the surface from the sky and surroundings, and the radiation emitted by a black body at outdoor air temperature. " ASHRAE Handbook suggests using $\Delta R \simeq 0$ for a vertical surface. ΔR for a horizontal surface is calculated by

$$\Delta R = \sigma(T^4_{sky} - T^4_{amb}) \qquad (kJ/m^2\ hr\ °C)$$

Infrared radiation is accounted for by a radiation heat transfer coefficient, h_{ir}.

$$h_{ir} = 4\sigma\epsilon_w \bar{T}^3 \qquad (kJ/m^2\ hr\ ^oC)$$

The total heat transfer coefficient for convection and radiation is :

$$h_o = h_w + h_{ir} \qquad (kJ/m^2\ hr\ ^oC)$$

Finally, the various heat transfer rates are given by :

$$\dot{Q}_{wall,cond} = Aq''_o$$

$$\dot{Q}_{window,cond} = UF_w A(t_{amb} - t_{rc})$$

$$\dot{Q}_{SHG} = F_w(1 - F_s)\ \bar{\tau} A H_T$$

$$\dot{Q}_c = \dot{Q}_{wall,cond} + \dot{Q}_{window,cond}$$

$$\dot{Q}_T = \dot{Q}_c + \dot{Q}_{SHG}$$

where U = overall heat transfer coefficient of window (U = 21.7, 12.3 and 8.2 $(kJ/m^2\ hr\ ^oC)$ for window with 1, 2 and 3 glazings, respectively). It should be emphasized that the correct number of inputs and parameters must be supplied for each mode.

Pitched Roof and Attic

Because of inevitable geometric assumptions about the roof and attic, all transfer function coefficients required for this model are stored for immediate access by simply specifying the thickness of ceiling insulation. FIGURE B.12.1 shows the assumed roof configuration and the four optional ceiling constructions.

Because empirical equations have been built into this model, only the units specified for each variable in the nomenclature may be used.

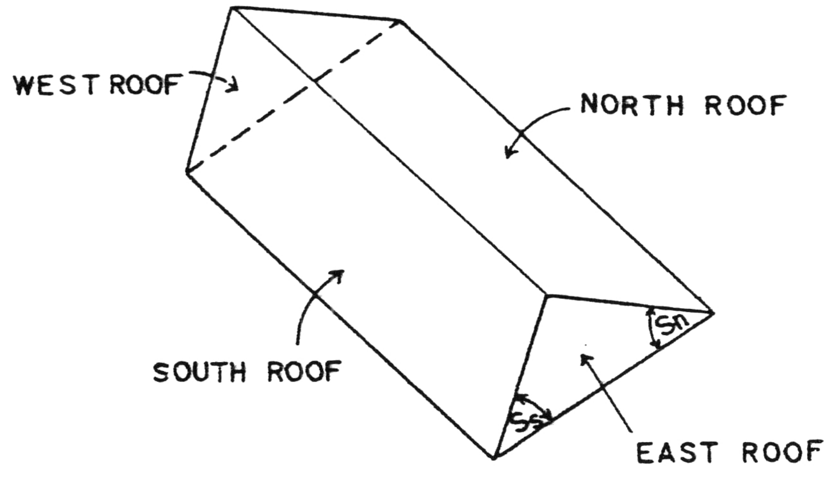


FIGURE B.12.1 (a) ROOF CONFIGURATION

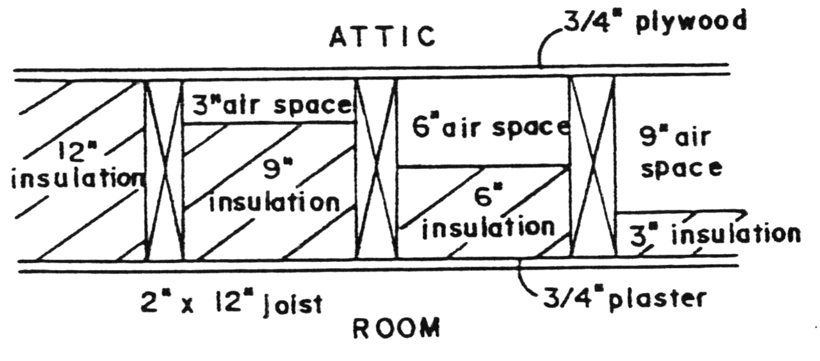


FIGURE B.12·1 (b) VARIOUS TYPES OF CEILING CROSS SECTION

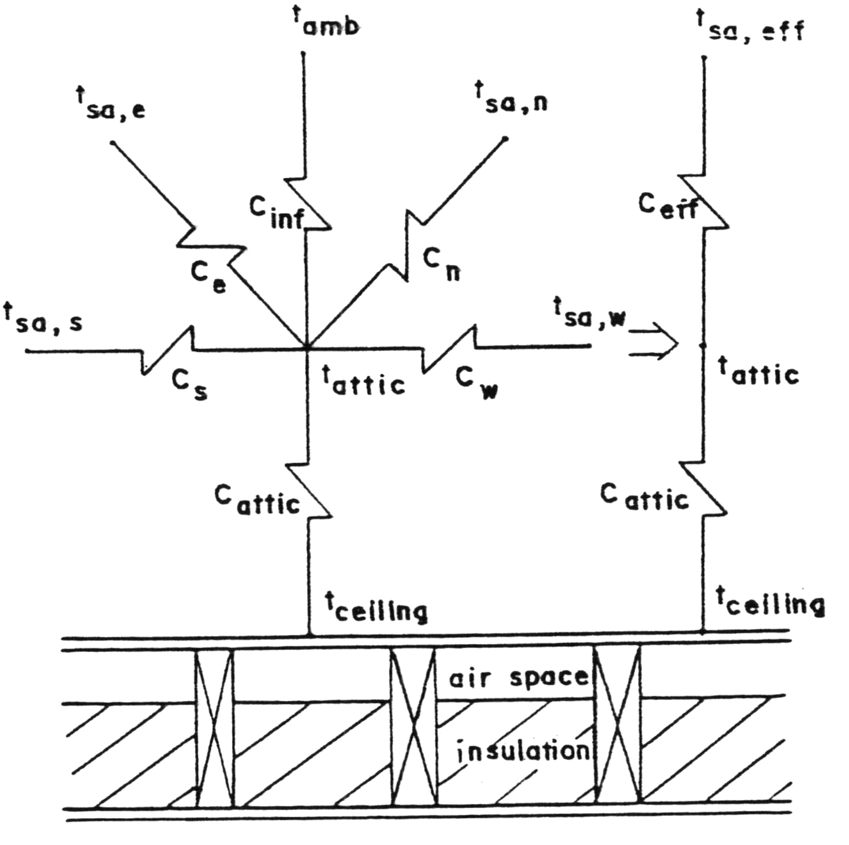


FIGURE B.12.2 EFFECTIVE SOLAR-AIR NETWORKS

Let

A_c : total ceiling area (m^2)

A_i : area of roof surface i (m^2)

b_n : transfer function coefficient based on a temperature difference term at time n (Btu/hr ft^2 ^oF)

C_{eff} : effective conductance from temperature $t_{sa,eff}$ to t_{attic} (kJ/hr ^oC)

C_i : heat conductance from roof surface i to t_{attic} (kJ/hr ^oC)

C_I : effective heat conductance of infiltration air (kJ/hr ^oC)

c_p : specific heat of air (kJ/kg ^oC)

d_n : transfer function coefficient for heat flux term at time n (-)

$F_{i,sky}$: view factor from roof surface i to sky (-)

$F_{i,grd}$: view factor from roof surface i to ground and surroundings (-)

$h_{i,s}$: inside heat transfer coefficient by convection for surface at slope s (kJ/m^2 hr ^oC)

h_o : heat transfer coefficient by radiation and convection at the outer surface (kJ/m^2 hr ^oC)

H_{T_i} : solar radiation flux incident on roof surface i (kJ/m^2 hr)

N_{roof} : parameter indicating whether a solar collector is attached to the roof (N_{roof} = 1) or not

(N_{roof} = -1) (-)

N_{ins} : parameter specifying the thickness of the insulation in Figure B.12.1 (b)
(N_{ins} = 1, 2, 3, 4 for 3", 6", 9" and 12" ceiling insulation, respectively) (-)

ΔR : infrared radiation difference between surface, and sky and surrounding (kJ/hr m^2)

q''_o : current heat flux entering (or leaving) a room (kJ/m^2 hr)

q''_n : heat flux entering (or leaving) a room at time n (kJ/m^2 hr)

$\dot{Q}_o$: total rate of heat gain (or loss) through the ceiling (kJ/hr)

S_i : slope of roof surface i from horizontal (^{o})

T_{amb} : ambient absolute temperature

t_{amb} : ambient temperature (^oC)

t_c : outlet fluid temperature from solar collector, if one exists (^oC)

$t_{sa,i}$: solar-air temperature of surface i (^oC)

T_{sky} : sky absolute temperature (^oK)

T_{grd} : absolute temperature of ground and surroundings (^oK)

$t_{sa,eff}$: effective solar air temperature of roof surfaces and infiltration air (^oC)

$\dot{V}$: rate of outside air infiltration of attic (m^3/hr)

W : wind speed (m/sec)

F_c : fraction of south roof that is solar collector area (-)

α : solar absorptance of exterior roof surface ($0 \leq \alpha \leq 1$)

ϵ_i : infrared emmitance of surface i ($0 \leq \epsilon_i \leq 1$)

σ : Stefan-Boltzmann constant ($\sigma = 2.041 \times 10^{-7}$ kJ/m^2 hr $^oK^4$)

ρ : density of outside air (kg/m^3)

The following description is based on that of TRNSYS.[2] An effective solar-air temperature, $t_{sa,eff}$, is calculated as a function of the solar-air temperatures of the various roof surfaces and the air infiltration rate. Then $t_{sa,eff}$ is used as the driving force in the heat flux equation.

$$t_{sa,eff} = \frac{C_s t_{sa,s} + C_e t_{sa,e} + C_n t_{sa,n} + C_w t_{sa,w} + C_I t_{amb}}{C_{eff}} \quad (^oC)$$

where the effective conductance, C_{eff}, is defined as

$$C_{eff} = C_s + C_e + C_n + C_w + C_I \quad (kJ/hr\ ^oC)$$

$$C_I = \dot{V} \rho c_p \quad (kJ/hr\ ^oC)$$

The solar-air temperature, $t_{sa,i}$, of surface i is computed from

$$t_{sa,i} = t_{amb} + \frac{\alpha H_{T_i}}{h_o} + \frac{\Delta R_i}{h_o}$$

where

$$\Delta R_i = \epsilon_i \sigma \left[F_{i,sky}(T^4_{sky} - T^4_{amb}) + F_{i,grd}(T^4_{grd} - T^4_{amb})\right]$$

and $\Delta R_{east} = \Delta R_{west} \simeq 0$

The view factors from surface i are, respectively:

$$F_{i,sky} = ½(1 - \cos(S_i))$$

$$F_{i,grd} = ½(1 - \cos(S_i))$$

The inside film heat transfer coefficient is a function of the slope of the surface. According to ASHRAE Handbook,[12]

$$h_{i,90^o} = 29.84 \quad (kJ/m^2\ hr\ ^oC)$$

$$h_{i,45^o} = 32.70 \quad (kJ/m^2\ hr\ ^oC)$$

$$h_{i,0^o} = 33.32 \quad (kJ/m^2\ hr\ ^oC)$$

The current heat flux into (or out of) the room through the ceiling, q''_o, is given by

$$q''_o = \sum_{n=0} b_n(t_{sa,eff,n} - t_{rc}) - \sum_{n=1} d_n q''_n \quad (kJ/m^2\ hr)$$

where b_n and d_n are transfer function coefficients. Finally the total rate of heat gain through the ceiling is

$$\dot{Q}_o = A_c q''_o \quad (kJ/hr)$$

Room and Basement

The model takes the energy balance of a room and basement using an ordinary differential equation. Heat flow through the basement (or slab floor), infiltration heat losses, and internal heat generation from room occupants, lights, appliances, etc. are accounted for in this model. Heat flows into (or out of) the room through the walls, the ceiling and other mechanisms, such as heat loss from the storage tank, can easily be introduced as external heat loads.

The time distribution of room heat gains has been built into this model according to the 1972 ASHRAE Handbook of Fundamentals[12] using the transfer function method. It also states that, in most cases, a slight fraction of the heat input is lost to the surroundings without contributing the heating or cooling load. This fraction is generally a function of the thermal conductance between the room air and the surroundings. If desired, this effect may be introduced into the model through parameter IFC. However, this effect is deactivated by setting IFC = 0.

For convenience, the model also incorporates a heat exchange with constant heat transfer effectiveness, to provide space heating and cooling. The exchange can be disabled by simply setting its mass flow rate to be zero.

As in the last two models, only the units specified by the nomenclature for each variable or parameter may be used.

Let

AREA : floor area of room (m^2)

CAPAC : thermal capacitance of room $(kJ/^oC)$

C_{min} : smaller thermal capacitance of the two flow streams entering the load heat exchanger $(kJ/hr\ ^oC)$

c_p : specific heat of fluid entering load heat exchanger $(kJ/kg\ ^oC)$

DEPTH : height of basement walls, if there is a basement (m)

EFF : constant heat transfer effectiveness of load heat exchanger (-)

F_c : fraction of heat gain that actually contribute to space heating or cooling load (-)

IBASE : a parameter indicating whether the room has a basement (-) (IBASE = 1 if a basement exists; 0 if no basement and no heat loss through floor; -1 if no basement but with heat loss through slab floor)

$\dot{M}_{in}$: mass flow rate entering load heat exchanger (kg/hr)

N : total no. of input variables (-)

PERIM : length of exposed room perimeter (m)

PEPL : number of room occupants (-)

$\dot{Q}_{base}$: rate of heat transfer into the room through the floor (kJ/hr)

$\dot{Q}_{bsfr}$: rate of heat transfer into basement through basement floor (kJ/hr)

$\dot{Q}_{bswl}$: rate of heat transfer into basement through basement walls (kJ/hr)

$\dot{Q}_{exs}$: net rate of heat gain into the room (kJ/hr)

$\dot{Q}_{gen}$: rate of heat generation by lights and electrical appliances (kJ/hr)

$\dot{Q}_{infl}$: rate of heat transfer into the room by air infiltration (kJ/hr)

$\dot{Q}_{load}$: instantaneous heating or cooling load (kJ/hr)

$\dot{Q}_{tran}$: rate of heat transfer into the room through load heat exchanger (kJ/hr)

RATE : number of air changes per hour because of infiltration and ventilation (1/hr)

T_{room} : room temperature (^oC)

T_{in} : temperature of fluid entering load heat exchanger (^oC)

T_{out} : temperature of fluid leaving load heat exchanger (^oC)

T_{amb} : ambient air temperature (^oC)

T_{grd} : ground water temperature around the house (^oC)

UA : overall heat loss coefficient of room (kJ/hr ^oC)

VOL : volume of room (m^3)

v_i : room transfer function coefficient (-)

w_i : room transfer function coefficient (-)

The following model description is based on TRNSYS[2]

$$\dot{Q}_{infl} = (RATE)\,(VOL)\,(1.2185)\,(T_{amb} - T_{room})$$

$$\dot{Q}_{light} = \dot{Q}_{gen} + (\dot{Q}_{light})_{input}$$

$$\dot{Q}_{pepl} = (237)\,(PEPL) + (\dot{Q}_{pepl})_{input}$$

where 1.2185 (kJ/kg ^oC) is the specific heat of air; $\dot{Q}_{light}$ is the total rate of heat gain from light sources and electric appliances; $(\dot{Q}_{light})_{input}$ is the time-dependent rate of heat gain from light sources introduced as an input; $\dot{Q}_{pepl}$ is the total rate of heat gain from room occupants; 237 (kJ/hr person) is approximately 225 (Btu/hr person), the value suggested by ASHRAE[12]; and $(\dot{Q}_{pepl})_{input}$ is the time-dependent rate of heat input by occupants introduced as an input. The loads due to solar heat gain, $\dot{Q}_{SHG}$, and conduction heat gain through walls, $\dot{Q}_{cond}$, are also introduced as inputs to the model. If an input load, say $(\dot{Q}_{pepl})_{input}$, is inapplicable, simply set it equal to zero during the simulation.

The loads due to solar heat gain, conduction, light sources, and room occupants are time-dependent by the same transfer function that computes the current heat load as a function of previous heat loads, current heat gain and previous heat gains.

$$\dot{Q}^t_{o,j} = \sum_{i=0}^{3} v_i \dot{Q}_{i,j} - \sum_{i=1}^{3} w_i \dot{Q}^t_{i,j}$$

where $\dot{Q}^t_{i,j}$ is heat load at i previous time steps; $\dot{Q}_{i,j}$ is the rate of heat gain at i previous time steps; v_i and w_i are coefficients specified by the construction weight parameter; and, j = SHG for solar heat gain, j = cond for conduction heat gain, etc.. The values of v_i and w_i will automatically be modified by multiplication with F_c when IFC = 1. Here

$$F_c = f(K_T)$$

where

$$K_T = (UA)/(PERIM)$$

See Chapter 22 of 1972 ASHRAE Handbook of Fundamentals(12) for further detail on the calculation of F_c.

Obviously, the current total time-distributed heat load is the sum of the four current time-distributed heat loads ($\sum_{j=1}^{4} \dot{Q}^t_{o,j}$). Other instantaneous heat loads, such as latent heat effects, are introduced into the model as inputs. Furthermore, $\dot{Q}_{base} = 0$ if IBASE = 0. If IBASE = 1,

$$\dot{Q}_{base} = \dot{Q}_{bswl} + \dot{Q}_{bsfr}$$

where

$$\dot{Q}_{bswl} = (PERIM)\,(DEPTH)\,(1.136)\,(0.5(T_{grd} + T_{amb}) - T_{room})$$

$$\dot{Q}_{bsfr} = (AREA)\,(1.136)\,(T_{grd} - T_{room})$$

If IBASE = -1,

$$\dot{Q}_{base} = (PERIM)\,(2.1633)\,(T_{amb} - T_{room})$$

The heat transfer rate through load heat exchanger is

$$\dot{Q}_{tran} = \begin{cases} 0 & \text{if } \dot{M}_{in} \doteq 0 \\ (C_{min})\,(EFF)\,(T_{in} - T_{room}) & \text{if } \dot{M}_{in} > 0 \end{cases}$$

$$T_{out} = T_{in} - \frac{\dot{Q}_{tran}}{(\dot{M}_{in})\,(c_p)}$$

Obviously, the current total instantaneous and time-distributed load, $\dot{Q}_{load}$, is the sum of all the appropriate heat loads mentioned above. Then, the net rate of heat gain into the room is

$$\dot{Q}_{exs} = \dot{Q}_{load} + \dot{Q}_{tran}$$

Finally, an energy balance on the room gives

$$\frac{dT_{room}}{dt} = \dot{Q}_{exs}/(CAPAC)$$

The above differential equation can be solved simultaneously with other system equations to simulate the behavior of the room temperature.

APPENDIX C

PARAMETER ESTIMATION

This appendix shows how some of the more important parameters are estimated.

Unit No. 5 (Case 1):

Since $$U \cong \frac{k}{L}$$

where $k = 0.1794 \ \text{kJ/hr m} \ ^{o}C$

$L = 26.0 \ \text{cm}$

Therefore,

$$U = \frac{0.1794}{0.26} = 0.69 \ \text{kJ/hr m}^2 \ ^{o}C$$

and $$UA_1 = 0.69 \times (2+4)$$

$$= 4.14 \ \text{kJ/hr} \ ^{o}C$$

Similarly,

$$UA_2 = 0.69 \times 4$$

$$= 2.76 \ \text{kJ/hr} \ ^{o}C$$

and UA_3 is equal to UA_1.

Unit No. 16 (Case 1):

U_{BE} can be calculated from

$$U_{BE} = U_b + U_e$$

$$U_b \cong \frac{k}{L}$$

$$= \frac{0.1794}{0.05}$$

$$= 3.58 \quad kJ/hr\ m^2\ {}^oC$$

$$U_e = \frac{U_{edge} A_{edge}}{A_{collector}}$$

$$= \frac{0.1794 \times (10+10+4+4) \times 0.095}{0.05 \times 80}$$

$$= 0.12 \quad kJ/hr\ m^2\ {}^oC$$

Therefore, $U_{BE} = 3.58 + 0.12$

$$= 3.70 \quad kJ/hr\ m^2\ {}^oC$$

Unit No. 17:

CAPAC is estimated by

$$CAPAC \cong C_{furniture}$$

From Chapman,(20),

$$C_i = V_i \rho c_p$$

$$C_{furniture} = \left(V \rho c_p\right)_{furniture}$$

$$V \cong 5\% \text{ of room}$$

$$= 15.95 \quad m^3$$

$$\rho_{oak} = 608.68 \quad kg/m^3$$

$$c_p = 2.3865 \quad kJ/kg/^oC$$

$$C_{furniture} = 15.95 \times 608.68 \times 2.3865$$

$$= 23169.2 \quad kJ/^oC$$

Therefore, CAPAC $\cong$ 23000 kJ/^oC

UA is estimated by

$$UA \cong (UA)_{furniture} + (UA)_{windows} + (UA)_{walls} + (UA)_{ceiling} + (UA)_{floor}$$

$$U_t = \frac{1}{R_1 + R_2 + R_3 + \ldots + R_n}$$

$$U_{furniture} = \left(\frac{1}{h_{r,f-ceiling}} + \frac{1}{h_w + h_{r,c-s}} \right)^{-1}$$

$h_{r,f-ceiling}$ (furniture to ceiling)

$$h_{r,f-ceiling} = \frac{\sigma(T_f^2 + T_c^2)(T_f + T_c)}{\left(\frac{1}{\epsilon_f}\right) + \left(\frac{1}{\epsilon_c}\right) - 1}$$

σ = Stefan-Boltzmann constant = 5.6697×10^{-8} W/m^2 $^oK^4$

T_f = furniture temperature = 23.2 ^oC or 296.36 ^oK

T_c = ceiling temperature = 296.36 ^oK

ϵ_f = emittance of furniture = ϵ_{wood} = 0.04628

ϵ_c = emittance of ceiling = 0.92

Therefore,

$$h_{r,f\text{-}ceiling} = \frac{5.6697\text{x}10^{-8}\ (296.36)^2+(296.36)^2\ 296.36+296.36}{\frac{1}{0.04628}+\frac{1}{0.92}-1}$$

$$= 0.272\ \ W/m^2\ {}^oK$$

Similarly,

$h_{r,c-s}$ $\cong$ 5.43 $W/m^2\ {}^oK$
(ceiling to surroundings)

h_w = 5.7 $W/m^2\ {}^oK$ for no wind

$$U_{furniture} = \left(\frac{1}{0.272}+\frac{1}{5.7+5.43}\right)^{-1}$$

$$= 0.266\ \ W/m^2\ {}^oC$$

$V_{furniture} \cong 15.95\ \ m^3$

If the average furniture thickness is 2 inches, then

$A_{furniture} \cong 314\ \ m^2$

Therefore, $(UA)_{furniture} = 0.266 \times 3.6 \times 314\ \ kJ/hr\ {}^oC$

$\cong 300\ \ kJ/hr\ {}^oC$

Similarly, we have

$(UA)_{windows} \cong 700\ \ kJ/hr\ {}^oC$

$(UA)_{ceiling} \cong 1770\ \ kJ/hr\ {}^oC$

$(UA)_{walls} \cong 335\ \ kJ/hr\ {}^oC$

$(UA)_{floor} \cong 600\ \ kJ/hr\ {}^oC$

$$UA = 300 + 700 + 1770 + 335 + 600 = 3700\ \ kJ/hr\ {}^oC$$

Functions QPEPL, QLIGHT and HWATER

Function QPEPL is constructed to describe the activities of the four occupants in the house. Two of them go to work in the morning and return home in the evening. The other two stay home most of the time. FIGURE C.1 shows the body heat generated by the occupants diurnally as a function of time. It is normalized so that the total heat is equal to unity. To obtain the amount of heat generation within any interval of time, we multiply the corresponding area under the curve of FIGURE C.1 by 18486 kJ, the average total heat generation by occupants within a day.

Like QPEPL, function QLIGHT represents the diurnal heat loads generated by the lights and electric appliances listed in TABLE C.1. The heat loads are estimated from the average hours and number of occupants in the house within each day. FIGURE C.2 shows the normalized diurnal electrical heat loads as a function of time. As above, we multiply the value read from FIGURE C.2 by 29673 kJ (the average total heat generation by lights and electric appliances within a day) to obtain the instantaneous electrical heat load.

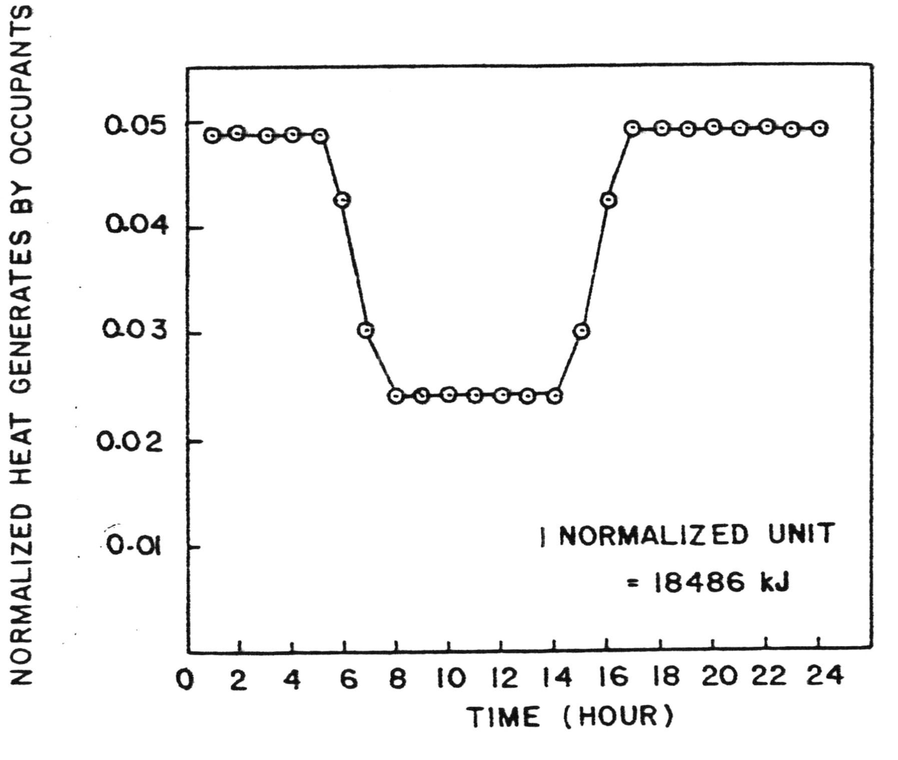


FIGURE C.1 HEAT GENERATES BY OCCUPANTS

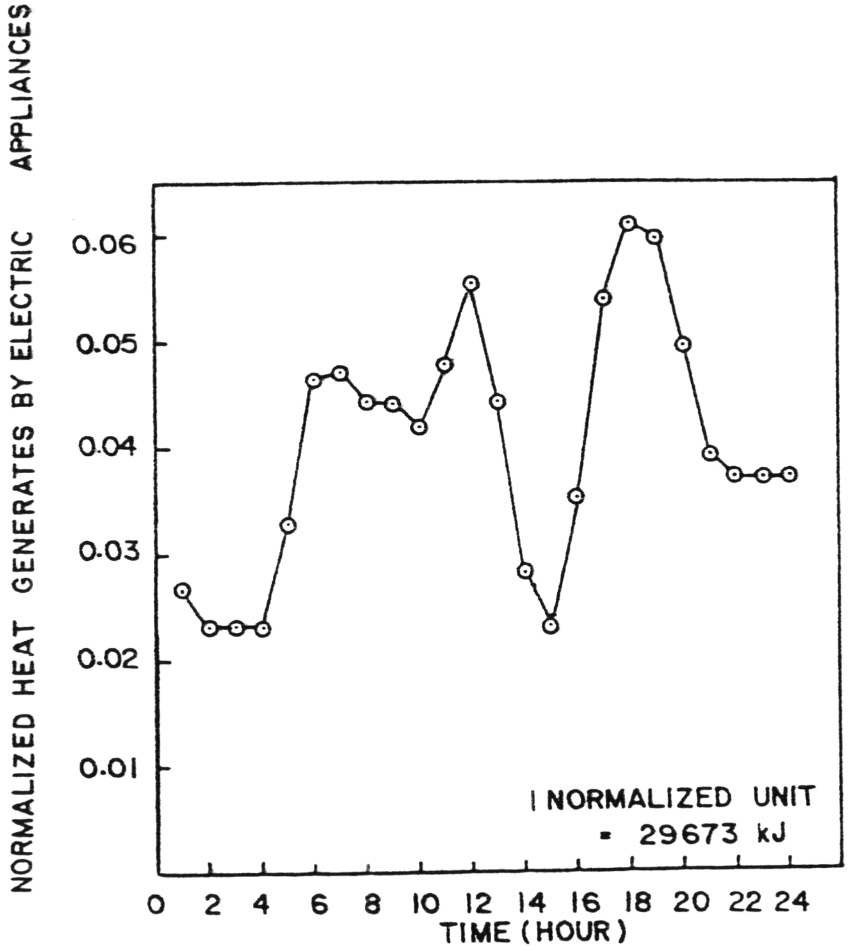


FIGURE C.2 HEAT GENERATES BY ELECTRIC APPLIANCES

TABLE C.1

List of Electric Appliances in the House

Name of Electric Appliances	Rating (Watt)
1. Radio/recorder stereo	109
2. Color television 20"	79
3. Refrigerator automatic deforst 5 cu. ft.	200
4. Rice cooker (2 liters)	600
5. Iron	750
6. Fluorescent lamp 10 units each	20

Similarly, function HWATER[34] predicts the diurnal hot water demands. The total daily demand is 1059.8 kg or approximately 1.06 m^3. FIGURE C.3 shows the normalized diurnal domestic hot water demand.

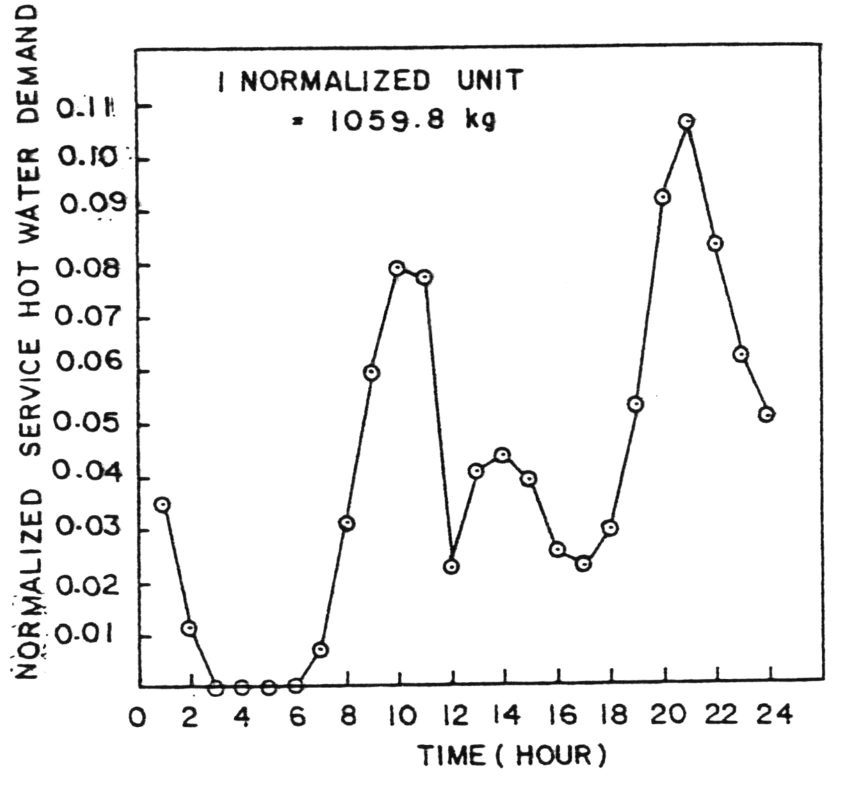


FIGURE C.3 SERVICE HOT WATER DEMAND

VITA

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